TREBALL FI DE GRAU

Grau en Enginyeria de la Energia

DATA CENTER ENERGY MODULES

Memòria i Annexos

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Resum

L'objectiu d'aquest document és el dimensionament d'un centre de dades amb reutilització de la calor residual, centrant-se en el disseny modular de las seves diferents parts, el funcionament del sistema de refrigeració fent servir CO_2 com a refrigerant principal i l'anàlisis de les conseqüències energètiques que es dedueixen, tot elaborat a partir d'una revisió de literatura prèvia sobre les diferents tecnologies aplicades. Aquesta literatura és precisament el punt d'inici del disseny ja que permet reunir tota la informació necessària sobre els centres de dades, la maquinària de refrigeració... per posteriorment, desenvolupar una idea pròpia de l'arquitectura i els seus modes d'operació referents a l'energia tèrmica. La idea de dissenyar un centre de dades de baix consum energètic comporta a analitzar els resultats al voltant de paràmetres com el PUE (Power Usage Effectiveness, Efectivitat de la Potència Usada en català), que és una mesura del consum elèctric d'un centre de dades, o l'estalvi energètic que deriva de la utilització de la calor residual. La principal conclusió extreta d'aquesta metodologia de disseny de centres de dades és que té beneficis en tots els àmbits. L'estructuració modular i enginyeril redueix costos a la companyia i condueix a guanys en l'eficiència, degut a un millor ús dels fluxos tèrmics. A més, l'impacte ambiental de la reutilització de calor i les positives conseqüències econòmiques per el consumidor proporcionen a aquest tipus de projectes una bona perspectiva de futur.

Resumen

El objetivo de este documento es el dimensionamiento de un centro de datos con reutilización del calor residual, centrándose en el diseño modular de sus diferentes partes, el funcionamiento del sistema de refrigeración usando CO_2 como refrigerante principal y el análisis de las consecuencias energéticas que derivan, todo ello elaborado a partir de una revisión de literatura previa sobre las diferentes tecnologías usadas. Esta literatura es precisamente el punto de inicio del diseño ya que permite reunir toda la información necesaria sobre los centros de datos, sus maquinaria de refrigeración... para posteriormente desenvolver una idea propia de su arquitectura y sus modos de operación en lo que a refrigeración se refiere. Esta idea de diseñar un centro de datos de bajo consumo energético lleva a analizar los resultados en torno a parámetros como el PUE (Power Usage Effectivenes, Efectividad de la Potencia Usada en castellano), que es una medida del consumo eléctrico de un centro de datos, o el ahorro energético que deriva de la utilización del calor residual. La principal conclusión extraída de esta metodología de diseño de centros de datos es que existen beneficios en todos los ámbitos. El diseño modular e ingenieril reduce costes a la compañía y conduce a ganancias en la eficiencia, debido a un mejor uso de los flujos energéticos. Además, el impacto ambiental de la reutilización de calor y las positivas consecuencias económicas para el consumidor proporcionan a este tipo de proyectos buena perspectiva de futuro.

Abstract

An approach of the sizing of a data center that reuses the waste heat is the aim of this document, focusing in the modular design of its different parts, the perform of the cooling systems using CO_2 as main coolant and an analysis of the energetic consequences extracted from it, all together with a previous literature review of every technology used. This literature review was actually the starting point of the design as it allowed to gather information about data center, its cooling machinery... for later develop an own idea of the arquitecture and the thermal energy operation of the designing. Due to this idea of building a low energy consumption data center, the results were analyzed around parameters like the PUE (Power Usage Effectivenes), which is a measure of the electricity consumed by a data center, and the energy savings derivated of the waste heat management cooling systems. The main conclusions that are extracted from this way of designing that centers, is that there are benefits in every aspect studied. A modular pre-engineerized structuring reduces cost for the company and leads to efficiency gains, as all energy flow is better utilized. Furthermore, the environmental impact of the waste heat usage and its positive economical consequences gives to this type of projects very good future perspective.

Acknowledgement

I would first like to thank my project advisor Armin Hafner from the Department of Energy and Process Engineering at Norwegian University of Science and Technology for offering me of the opportunity of perform this project. He consistently allowed this paper to be my own work, but steered me in the right the direction whenever he thought I needed it.

I would also like to thank the postdoctoral fellow Ángel Álvarez Pardiñas for its good guidance when some aspects of the project were not totally understood or when information about the design were needed.

Índex

1. INTRODUCTION

1.1. Goals of the project

Due to the growth in data centers number and power usage, it is needed to apply to them engineering solutions regarding size, location, flexibility, cooling options, etc. Thus, the main goal of this paper is to make an approach of the design of one of this IT installations, focusing in the development of a modular design and the cooling technologies that can be applied. The potential of this systems when it comes to energy supply are great, as it can be said that they give it for free. This systems consume important amounts of electricity as they are computers storing and processing data constantly, thus also need big cooling technologies to cool down all the servers they have. If new engineering solutions can manage all that waste heat properly and reuse it for different applications, this systems could just turn into a very important part of the thermal energy supply at world scale.

1.2. Scope of the project

This project is mainly an overview of the data center structure, energy flows and performance, as well as a review of the cooling technologies that usually appear in this kind of installations. The idea is not to develop a definitive or totally accurate solution, with exact measures or fix arquitecture of the modules, but giving a initial view of the potential of this infrastructures as energy supply systems and the necessity of making use of that amount of energy. In further related projects, many of the parameters set in this design could be modified and improved and more accurate calculations and simulations could be done, but the axis around which the idea is developed is already settled.

1.3. Software used

1.3.1. CoolPack

This is a very simple program developed by the DTU (Technical University of Denmark), updated the last time in 2012. This system allow to calculate directly almost any type of cooling cycle just by introducing some initial values that the software uses as reference. It provides with all the thermodynamic states of that are needed and also more information about the cycles like the mass flows, the COP's, the heat rejected and absorbed... Furthermore, it is possible to select Carnot cycles with different initial settings like simple ones, cascade systems, different evaporator and condenser options, etc. and it also gives the possibility to perform CO_2 transcritical cycles. In spite of its simplicity, this software allow to ease all the thermodynamic cycles calculations without using a lot of time.

1.3.2. HXSim 2007

This is a software developed by the Norwegian independent research organisation called SINTEF. The purpose of the program is to simulate the behaviour of heat exchangers performing under different circumstances. Thus, It is possible to design a heat exchanger since the beginning and it allows to modify almost every possible variable in this cooling devices in order to develop one that suits the needs. After introducing that initial amount of data, the simulator can be ran and, through an iterative method, finds out if that configuration is possible regarding pressure drop in the coolant or air side, proper size of the heat exchanger, acceptable tube diameter and several different parameters. Finally, the simulator provides different graph options and report in order to have an overview of the heat exchanger performance. It is not possible to have public access to this software, as it is only thought for being used by the organisation.

2. LITERATURE REVIEW

Several sources were reviewed and checked in order to be able to develop an idea that suits the demands. First of all, the actual shapes, structures and components of data centers were studied, as well as the heat load released by all the servers. Most of the actual ways of cooling data centres were studied with the purpose of knowing which are the strengths and weaknesses of each one and which would suit more for being applied in a modular way.

2.1. Data center consumption

Due to the growing amount of data that has to be stored by different companies or institutions, data center consumption has turned into a significant percentage of the energy needed by the most powerful countries in the world. In a global scale, from 2005 to 2010, data center consumption grew at a rate of 24%, showing a problem regarding control of the energy that this equipment was using [1], and recent studies claim that represents around the 1,3% of the energy consumption [2]. The biggest responsible of this situation are countries as U.S., China or Japan, which have even bigger shares than the value presented previously, and with remarkable growth rates [2].

Not only the amount of energy consumed by data centers is growing, but also the energy density of this IT installations, which means that the new servers have more power capacity using less volume. In 2013, the average power consumption per rack was 8 kW, but some studies claim that it will arrive to 52 kW per rack in 2025 [3]. There is a main inconvenient produced by this situation: the growing amount of waste heat the servers produce. This only can by solved by improving the cooling technologies and systems that are applied to data centers, which are starting to bear fruit as, according to [4], the growth in the energy consumption rate estimation for the period 2010-2020 is 4%, which represents a big decrease regarding the previous period, in spite of the growing data storage demand.

2.2. Data center structure

Data centers are becoming so common and important in the daily life of a lot of companies and institutions that many of the pieces of its structure and its measures have been standardized, in order to be able to produce them in a quick and flexible way. There are, consequently, parts in which data centers are divided

2.2.1. Server

A server is a computing device used for storing information and performing different kinds of actions when requested. This is the smaller part in which the data center is going to be divided. The first servers were rack-mounted, which means that they were directly place in a cabinet or rack and they had their own power supply system. Over time, the need of getting smaller and at the same time increase the storing capacity lead to develop rack-mounted servers that do not have their own power supply, as it is supplied by a PSU (Power Supply Unit) that as common for all the devices. This type of servers usually have 1U size (which is equivalent to 480 mm wide and 1,75 mm tall), being able to achieve 2U [2] depending on the number of processors.

Figure 2.1. Schematic view of the U unit an rack server (Source: [5] [6])

Finally These are a type of server able to storage the same or even more information in less space than a normal one in even less space, as its size is smaller than 1U. They are called blade servers and they are usually disposed in a enclosure.

Figura 2.2. Blade server (Source: [7])

2.2.2. Chassis

It is also called blade enclosure and it can contain up to 8 or 16 blades, although as the technology advance it is being possible to introduced more servers in the same space, even up to 180 [2]. The chassis also contains those elements that are general for every blade, such as the power supply, cooling system, networking, management... All of it in a space of 10U [2].

Figure 2.3. Blade server chassis (Source: [7])

The power supply is needed as the computers operate in ranges of DC voltages, providing then another source of heat to cool down. The PSU can be placed in different parts of the chassis and it usually takes 3U of space [2].

2.2.3. Rack

It is a metal structure that contains up to 64, 84 or 96 server blades [2],which means 4 or more chassis. Apart from the blades, it also contains the common elements for each server that are also contained by the chassis.

Usual measures of the structure are: 482,6 mm wide (19 inch), 42 U height, and 914,4 mm (36 inch) depth in the case of the known as 19-inch rack. However, depending on the fabricant, the brand or the requirements, there are a wide range of different measures that can be accepted.

Figure 2.4. Server rack with servers inside (Source: [8])

2.3. Thermal loads and temperature limits

In order to design a data center, it is important to know the amount of heat its being dealing with and the temperatures that, in general, the serves and devices can hold.

Taking a look into [2] it can be seen the main values regarding thermal loads and temperatures for the different elements of the data center, from the most to the less specific. All this information allow to develop table 2.1 which is a summary of all the information presented.

	Thermal load (W)	Temperature (ºC)
Processors	60-75	85
DIMM	6	85
Disk drive	12	45
Server	300-400	-
Rack	13000-26000	-

Table 2.1. Thermal values and temperature limits

Depending on how is the cooling system that is going to be designed for the data center, certain values are more important than others. For example, if the cooling is going to be liquid-base on-chip, then it is important to know the specific heat that every single device dissipates, whereas if the system is going to be air-based, knowing the total amount of heat that the rack releases is enough information for the design.

2.4. Cooling technologies in data centers

Once the thermal loads are known, it is important to investigate the different cooling methods and technologies that are being applied to data centers since their creation until nowadays, in order to know all the strengths and weaknesses of each one and develop the idea that suits the best the requirements. A lot of systems for cooling data centers are already existing or are being developed, so probably not all but most of possible solutions are shown then.

2.4.1. Computer Room Air Handler

There are several cooling technologies that can be applied to cool down a data center. The traditional way it is called CRAH/CRAC (Computer Room Air Handler) units. They use only air cooling supplied to the data room that comes either from a raised floor (there is a pendum floor and the air gets into the room by perforated tiles) or from a non-raised floor (where the aire is provided through diffusers in the ceiling).

Regarding the distribution in the room, this cooling technology requires that the racks are disposed face to face an back to back in order to create hot and cold isles. What happens is that the entrance air is cold so it stays in the low layers of the room, it goes through the racks to cool them down and then it rises due to stack effect. The distribution face to face and bottom to bottom helps not to mix the cold air with the warm one. This warm air is picked up by the HVAC system which takes it to be cooled down, usually in water chillers.

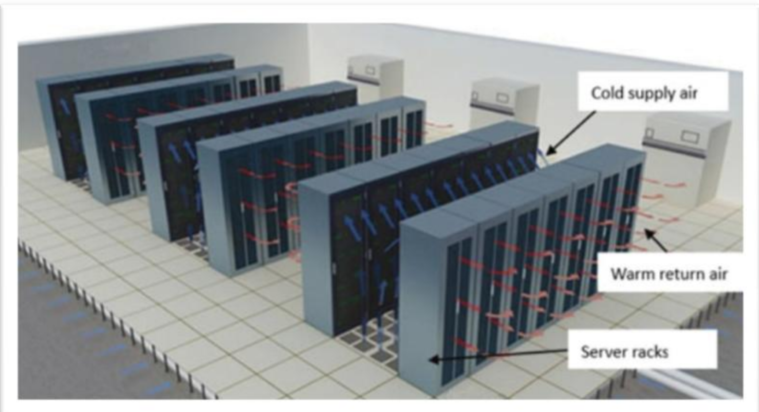

Figure 2.5. Computer Room Air Handler (Source: [1])

This systems have several cons according to [1]:

- The air has poor thermal properties. This leads to the need of increasing a lot air temperature and creates a big temperature difference within the server.
- There are some localized hot spots that need extra cool. For that, the temperature of the cooling air has to be extra low which is translated in more waste of energy and lower performance.
- The refrigeration system has to operate under the same conditions regardless of the outdoor climate.
- Heat generated is automatically removed and not reused.

Other shortcomings can be read in [2]:

- Ceiling height: Hot air stratification may occur.
- Airflow direction: The airflow recirculation is the biggest problems in terms of cooling. The temperature in the cold aisle is no uniform all along the rack, as in its top part, the inlet air temperature is increased, which decreases performance.

There have been several researches in the field of air cooling and numerical simulations are used for that purpose but the shortcomings of these systems are too big, so new technologies that increase energy efficiency are starting to be used.

2.4.2. Liquid cooling

As it is well known, liquids have better thermal properties than air, so using them is a expected direction of development regarding cooling of data centers. In this kind of systems there is no direct contact with the air as the cooling circuit is usually isolated. Basically, instead of being directly placed in the rack, the devices are set in cold plates which have tubes where the liquid is circulating and cooling down the different processors, disks... of the server. Regarding the liquid state, cold plates can be classified, following [1] in:

- Single phase cold plate: data centres mounted over a cold plate which acts as heat exchanger and transfers the heat by conduction to the pipes with coolant.
- Double phase cold plate: the process is similar to the previous one but in this case the coolant evaporates.
- The main differences between this two types are:
- The use of latent heat instead of sensible leads to a large cooling capacity.
- The use of latent heat creates a reduction in the coolant flow and more uniform cold.
- Vapour created when latent heat is used requires larger pipes. However, because of the buoyancy effect, it is possible to have a pumpless thermosyphon loop.

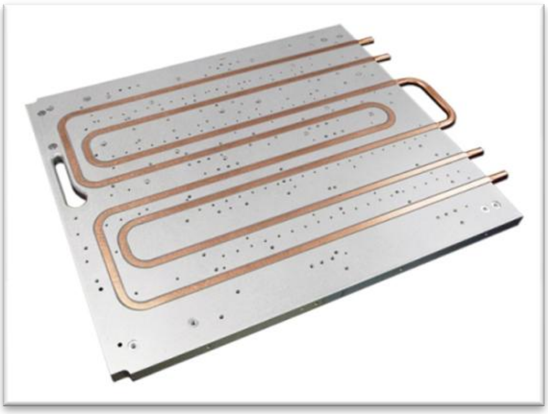

Figure 2.6. Cold plate (Source: [9])

As main strengths, cold plates have better performance that air cooled systems. The cooling is done by direct contact with the devices so handle hot spots is easier as they can be placed in a specific part of the plate which has more cool area, for example, whereas with air is not possible to have such a specific way of cooling. Furthermore, there is more control of the temperature as the variations of room heat gains are not affecting the system.

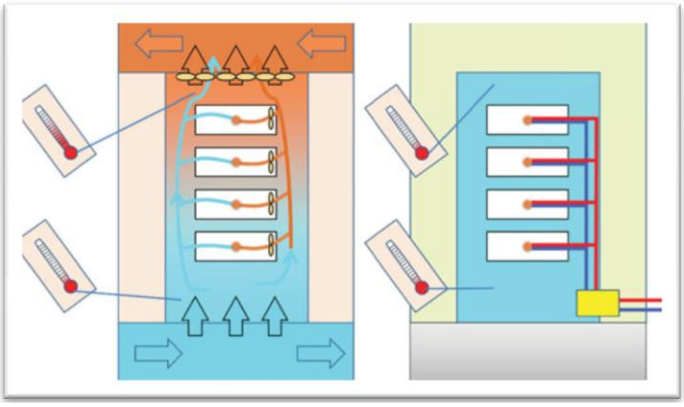

Figure 2.7. Comparison between air cooling and cold plate cooling (Source: [1])

However, this systems also have weaknesses. In spite of having better performance than the air cooled ones, they are also more expensive as its design requires more elements. Furthermore, it adds complexity when building the data centers. This fact leads to a loss of flexibility when a mass production of modules wants to be done and also can increase the time for installing them.

2.4.2.1. <u>On-chip liquid cooling</u>

Instead of installing the several devices on a cold plate, in this case they are cooled through a direct connection with the refrigerant. The energy savings can achieved a 60 % according to [10].

There are several ways of doing on-chip cooling, but the most attractive nowadays is the two-phase one.

Again, we can find the same weaknesses than in the previous system. Both the building and the installing process will increase, and maybe even more than in the prior situation, its difficulty as the tubes with coolant are directly installed in the specific device that wants to be cool down. The interesting point is that not all the elements of the server have to be installed in a cold plate so those that do not need such a specific cool can be cooled down by a traditional air cooling system.

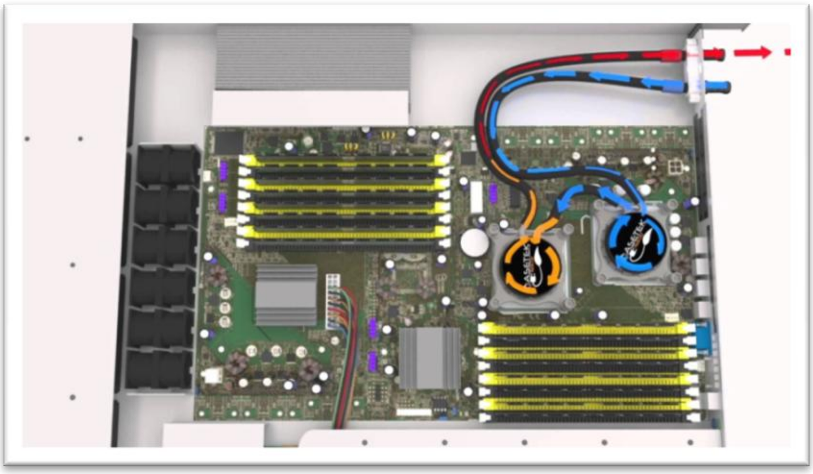

Figure 2.8. On-chip server cooling

2.4.2.2. <u>Immersion liquid cooling</u>

This system is probably the most innovative one of all the ones mentioned in this document. The idea is to introduce all the blades in a tank with a coolant oil inside. There is a pump that drives the coolant (it has to be a dielectric fluid as due to safety measures regarding the servers) and heat exchangers to connect with a second working fluid that removes the heat afterwards. Web page [11], a fabricant that works with this system, ensures up to a 95 % savings in the cooling consumption of the data centre. Even though it is an interesting technology, there is not much investigation around it and it is not very introduced in the market yet.

Figure 2.9. Immersion liquid cooling (Source: [11])

2.4.3. Other technologies and combinations

2.4.3.1. Coupled air and liquid

This system uses both water and air for cooling. Cold water is supplied and is used, in first place, to cool down the inlet air. This air is in charge of cooling some of the devices so there is no need to mount them on a cold plate. Afterwards, the water cools the rest of the devices placed on a cold plate and is finally driven to a cold water or air exchanger in order to cool it down.

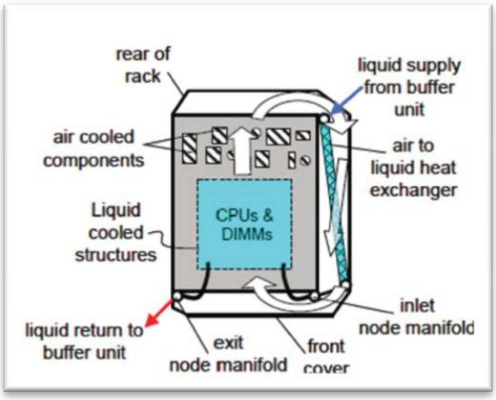

Figura 2.10. Air and liquid cooling system (Source: [1])

2.4.3.2. Free cooling

As outdoor temperature in Norway is usually cold enough for using free cooling, this can be a interesting option for cool down the data centre. For example, in [12] is developed a combined cooling system that works with liquid cooling and free cooling regarding the outdoor conditions. Moreover, in [13] and investigation of data centre free cooling using CO2 is developed. CO_2 is compared with refrigerant R22 (which will be forbidden due to its high GWP) and it is shown that R744 (how CO_2 is called when it works as refrigerant) shows a better performance than the R22 for this kind of applications . Using free cooling would mean energy savings due to the use reduction or removal of the CRAC system and the chiller.

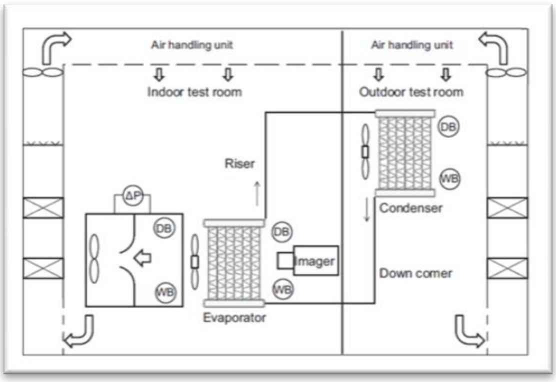

Figure 2.11. Scheme of a free cooling system for data center (Source: [1])

2.4.3.3. <u>Microchannels</u>

By using microchannels, the convective thermal resistance is reduced what increases the heat transfer to the coolant [1]. This technology can be applied either in a cold plate or directly in the different devices (on-chip). Even though on-chip cooling seems to be an interesting option, due to the existing possibility of leakages, some costumers don't accept it even using a dielectric liquid. Cold plates result to be safer.

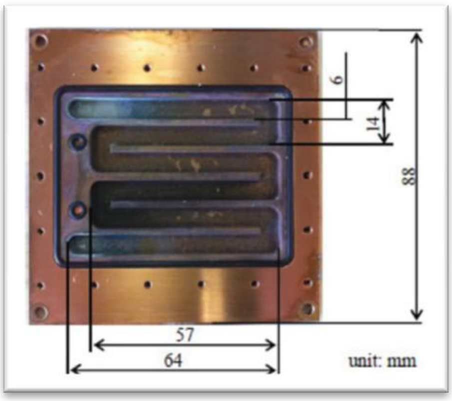

Figura 2.12. Microchannels cold plate (Source: [1])

2.4.3.4. <u>Heat pipe or thermosyphon</u>

There are three main ways in which heat pipes can be used for cooling according to [1], but two are mainly interesting.

- Heat-Pipe-Based air conditioning: This system works as a substitute of the CRAC one. It is basically a thermosyphon loop with gravity as driven force that extracts the heat from the data centre and rejects it to the outside. As advantage, outside air is not used directly so contaminants are avoided but the cons are similar to the CRAC system ones.

- Distributed Heat-Pipe system: In this case, cooling system is directly connected to the rack. The fans are placed at the top and the bottom of the structure and each one is connected to a heat pipe loop that cools down the inlet and the outlet air. This system avoids some of the shortcomings of the previous system, as there is no need in heating the room air for example. The hot spots problem, however, won't be solved with this system.

2.4.4. CO_2 as coolant

CO_2 was already being use as refrigerant during the Second World War, mainly when ammonia (NH_3) was considered a risk for the people because of its toxicity [14]. After this, synthetic refrigerants like HCFC's and HFC's were started to be used and CO_2 lost importance.

Nowadays, it starts to be used again because of climate change mitigation issues. Its Global Warming Potential is 1,which is very low and even more compared to the ones that the HCFC's and HFC have. In the beginning, its lower efficiency was a shortcoming to overcome but, known as being the only refrigerant that could help to reduce the global warming, several researches were done in how to adapt the cycles and its elements to improve energy efficiency, and it is showing that it can be a very interesting refrigerant for several appliances. It is registered in ASHRAE as R-744.

2.4.4.1. Thermal properties

As remarkable properties, CO_2 stands out first of all due to its low critical temperature. This is going to lead to a operation mode in the supercritical region which, as we will see later, will show some interesting facts about the refrigerant.

Its high triple point, situated at 5,18 bar, creates limitations in the suction pressure downwards this value and in the evaporation temperature, as it cannot be below the corresponding temperature of -56,6 ºC.

Finally, another interesting property is the high general pressure level. This fact has pros and cons as, for example, gives a high specific volumetric capacity (kJ/m^3) but requires specific components that can hold that pressure. Furthermore, as the saturation curve is quite steep, CO_2 presents low increments of temperature per increment of pressure, which is positive as the fact of working with high pressures does not imply to work with high temperatures. Figure 2.13 shows the phase diagram of CO_2 with some specifications.

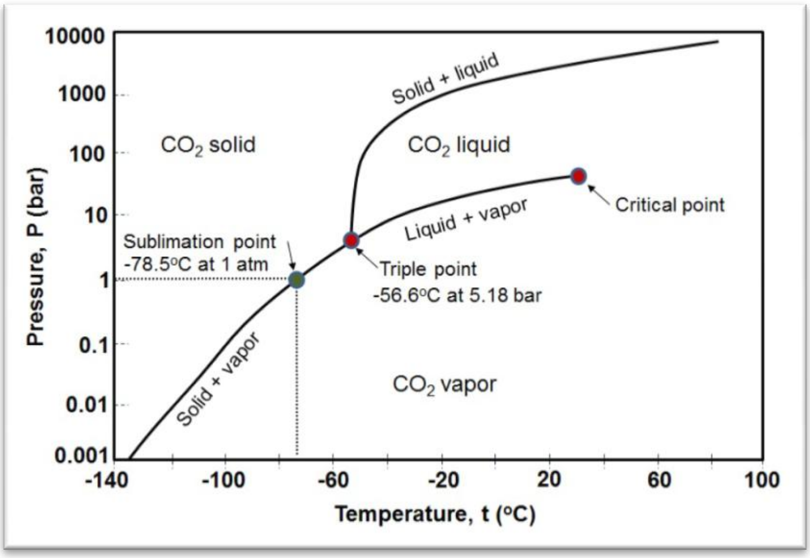

Figure 2.13. Diagram of the different states of CO_2 (Source: [14])

The table below [15] shows some CO_2 properties compared to other coolants commonly used.

Coolant	T_{critic} [ºC]	P_{critic} [bar]	Vaporization [kJ,kg], 0ºC	Molecular mass [uma]	GWP
CO_2	31,7	73,8	337,7	44	1
Ammonia	132	113,3	1261,7	17	0
R-134A	101,	40,6	198,4	102	1300
R-410A	49,5	49,5	221	73	1890

Table 2.2. Comparison parameters between several coolants

2.4.4.2. Transcritical cycle

This cycle started to be taken into account when CO_2 was becoming a important option when it comes to refrigerants. Several researches about this cycle showed that it can be very competitive regarding the classic coolants in several fields, mainly when the temperature glide in the warm side can be used for improving the cycles.

Transcritical cycle, however, is not beneficial and optimal for every situation.

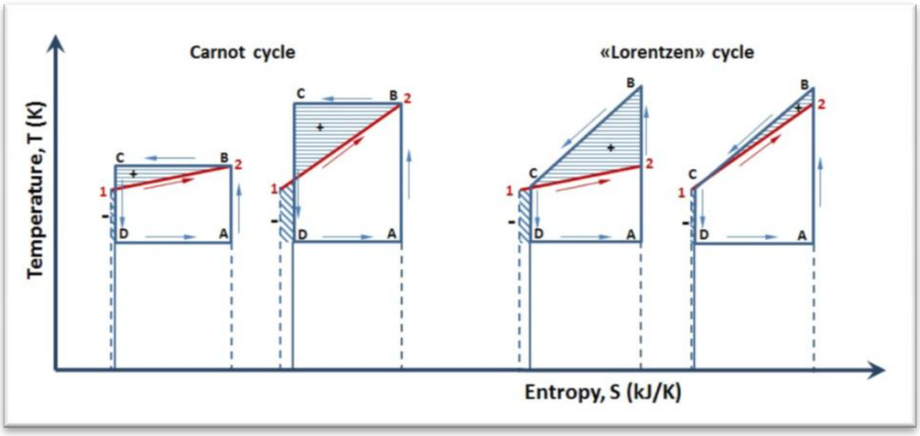

Figure 2.14. Comparision between Carnot and Lorentzen cycle(Source: [14])

Figure 2.14 shows some facts about the transcritical (Lorentzen) cycle in comparison with the classic Carnot cycle. When the temperature glide in the receiving working fluid (either in heating or cooling operation) is small, Carnot cycle shows very small losses due to temperature mismatch (Carnot cycle on the left), while Lorentzen cycle will give high losses and very small temperature match (the one on the left). In the opposite situation, when the temperature glide is high, Lorentzen cycle is able to adapt to properly to the temperatures (Loretzen on the right), so the mismatch is small and so are the losses, while the Carnot cycle shows big amount of losses due to poor temperature match (Carnot cycle on the right).

Regarding the specific properties of the CO_2 in the transcritical cycle, there are several points to stand out. When fluids are close to the critical point what is actually happening is that this fluid is going to pass from a state in which it has two different and separable phases to another state where there is an unique and continuous one. Thus, when a coolant is reaching this point and when is over it but close, its properties behave a lot like the ones of a fluid in phase change (with big changes in characteristics for a small temperature gradient). As the pressure keeps on rising and the critical point moves away, the fluid goes back to smooth changes in the properties.

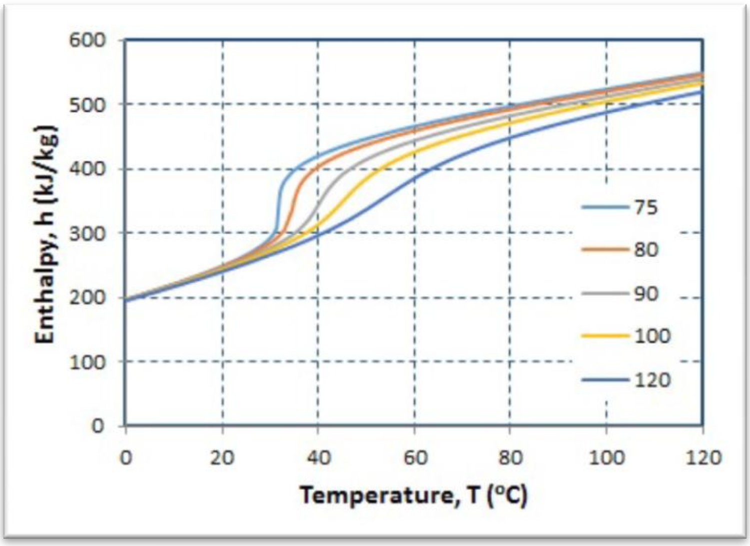

Figure 2.15. CO_2 enthalpy evolution in transcritical cycle (Source: [14])

Figure 2.15 shows well this fact. The CO_2 wants to be heated from 20 until 90ºC and, as it can be seen, it absorbs most of the heat in the range of temperatures between 25 and 40ºC. This is exactly the area mentioned before, a bit below and over the critical point. Afterwards, the curve goes back to a smooth increase of its properties.

Figure 2.16 shows the same idea, but with the T-h diagram. The part of the diagram close to the critical point shows a less steep curve. This basically means that, with the same temperature gradient, it is possible to get a bigger increase in the specific enthalpy which, in other words means that the fluid has better specific heat in that part of the diagram.

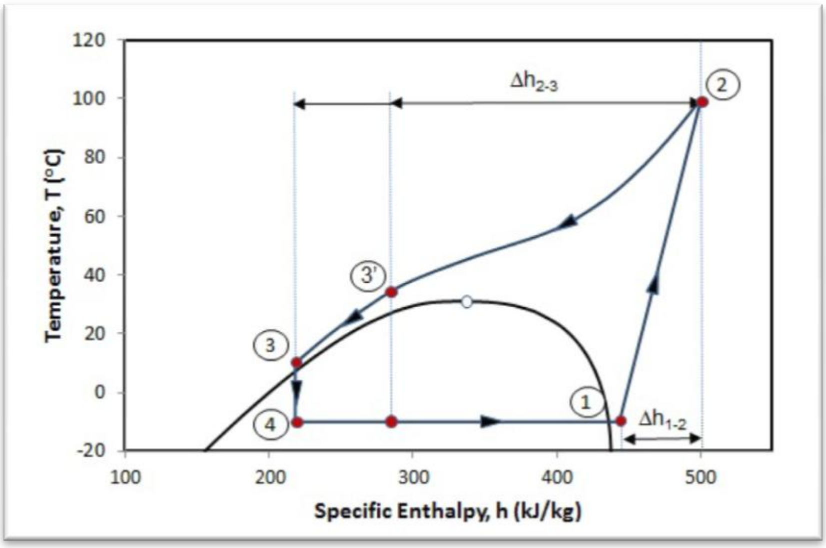

Figure 2.16. T-h diagram of Co_2 transcritical cycle (Source: [14])

CO_2 is good option to apply in transcritical cycles. As shown in table 2.2, it has a very low critical point so there is no need to add so much pressure to the cycle to reach the trancritical zone, which is an advantage in terms of pumping energy. Acting in a normal Carnot cycle, its operation will not be as good as, for example, ammonia due to low vaporization heat among other characteristics, and it would present a low Carnot performance. However, figure 2.17 shows that, in a Lorentzent cycle, CO_2 can reach very good COP values.

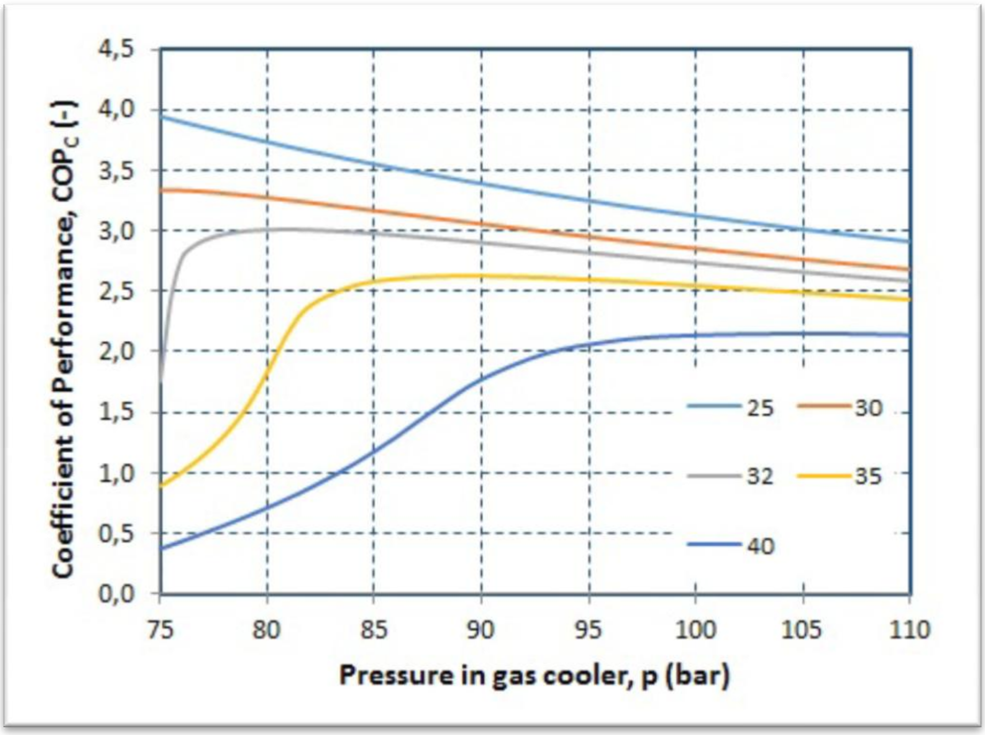

Figure 2.17. CO_2 transcritical cycle COP regarding gas cooler pressure (Source: [14])

2.4.5. Ammonia as coolant

Ammonia has been used since a long time as refrigerant due to its optimal thermal properties, as it can be seen in figure 2.18 compared to R-134A. Its vaporization enthalpy is great so there is a big amount of energy than it can absorbed when performing in a two phase system, that is why is usually considered in heat pumps with high capacities as 100 kW and above. Furthermore, it has a GWP of 0 which means that is a very ecological coolant as it does not contribute to the greenhouse effect.

	Ammonia	R-134a
Critical Temperature	133ºC	102ºC
Latent Heat at 80ºC	892 kJ/kg	106.4 kJ/kg
Latent Heat at 5ºC	1243 kJ/kg	193.4 kJ/kg
Pressure at 80ºC	41.4 bar abs	26.3 bar abs
Pressure at 5ºC	5.16 bar abs	3.50 bar abs
Pressure ratio	8.0	7.5
Vapor Quality after EV*	29.5%	60%
Isentropic Discharge T*	179.8ºC	92.3ºC
COP*	2.84	2.43

Figure 2.18. Comparative table between Ammonia and R-134A properties (Source: [16])

Ammonia, thus, have several applications nowadays in a lot of different industrial field. It substitutes coolants as R-11, R-12 or R-22 in actual projects that need refrigerant chilling. Ammonia chillers were installed in banks offices in the city of London for its cooling and similar system was installed in the International Maritime Organization (part of the UN). Very good energy performances and savings were reported after setting this devices. Furthermore, high efficiency free cooling systems have been working all together with ammonia as working fluid for cooling computer data centers for a long time and it has substitute R-134A in heat pump applications, as that one was the most used fluid working with heat pump technology. These are several examples of the very good performances ammonia show to have, and its uses has shown to be a very good solution for energy savings in many applications [16].

In spite of its good performances, not everything is perfect when working with ammonia. First of all, even though the risk is not high, ammonia is a poisoning fluid. It has a irritating odor that can be detected with small concentrations of 10ppm, even though the deadly concentrations are around 200 times higher. Furthermore, the fluid is flammable when mixing it with air with concentrations around 20 %, but those numbers are not easy to reach. If the professionals that work with ammonia are properly educated in safety measures, ammonia is not a dangerous coolant that moreover shows to be optimal regarding environmental and thermodynamic issues.

2.5. Modular design

Traditionally, data centers were placed in big infrastructures or rooms in buildings and there were no specific or previous design of them, so the disposition of the IT equipment and the cooling system was more like an open book, with no standards or prefabricated elements that facilitate the set up. However, over time, the scenario changed and modular designs started to be required.

As cloud computing was starting to be developed, it appeared the idea of transferring this concept also to the physical structure of the servers that hosted that cloud, the data center [17]. The thought is to develop not a specific data center, but a way of building them that gives physical advantages, as well as better energy performances and money savings.

This idea of growing in the direction of prefabricated and easy-to-install data centers lead to develop some of them that were, not only built in modular way, but also adapted to ISO shipping containers that provide even more flexibility. Usually, the concept of modular data center is confused with having containerized data centers [18]. The actual fact is that this second idea is involved in the first one, as "modular" is more like a concept, a way of building all the IT equipment and cooling systems in a manner that provides advantages, and containerized data centers refers to one of the ways of going modular, that does not have to be unique.

Either way, modular design of data centers offers a big amount of possibilities and facilities. In [19] some of them are well explained.

- Speed of deployment: Modular data centers have a small timeframe since they are ordered to the moment of deployment; a matter of months. Furthermore, the site construction can be develop in parallel regarding the manufacturing of the equipment.
- Scalability: As the design is standard, it is easy to adapt the demands to the offer, as different size modules can be used until finding the optimal solution. Moreover, this faculty facilitates the fact of substituting any obsolete module in a simple way or placing new ones in case of any disaster that can occur.
- Agility: Modular data centers are easy to adapt to any of the requirements than an specific company have. Either if it is preferred a more economic solution or one that stands out the performance, it is possible to develop a module that suits the needs.
- Mobility: Data centers designed in a modular way can be placed almost everywhere. They do not need big infrastructures and they are perfectly adaptable to the environment. They can even sometimes make use of the natural resources in the surroundings like, for example, for the cooling system. Furthermore, the fact of having modules facilitates the transport to different areas.
- Density and PUE: Modular solutions allow power densities of more than 20 kW per cabinet when traditional ones had typically around 100 W per square foot. Also modular design allow to have very low PUE (Power Usage Effectiveness) reaching values in the range of 1,1-1,4 (ideal value is 1).
- Efficiency: The fact of having such an integrated data center leads to best energy performances, regarding both power and cooling. The can be placed close to electric stations so the power losses and the energy distribution is minimized and,

furthermore, the waste heat produced and captured by the cooling module can be reuse for different applications as warming up water or district heating.

- Commissioning: Modular data centers can be commissioned in the place where they are built and in a very simple way.

It is also important to remark that the fact of building in a modular way leads to money savings, as the can be chain produced which reduces costs in operation and having standardized measures implies savings in materials acquisition.

3. THEORY BASE

3.1. Developed idea

The main concept of the idea developed is a data center that can be installed by modules. Data centers can be place in big warehouses where all the servers and the cooling systems are installed, but this configuration reduces highly the flexibility, as the system cannot be moved to a specific location of interest. The concept of modular data center is based on introducing all the elements that form this IT installations in containers that can be moved with different transport media to the location selected, for afterwards being place there in an efficient way. Furthermore, this idea tries to involve the use of CO2 as working fluid in the cooling system, due to its high latent heat capacity and its very low GWP, and the possibility of reusing the big amount of waste heat produced by the data center for different applications.

Several documents were checked in order to find out what would be the best option to develop the modular data center. A document [20] of the company TROX AITCS was found when looking for CO_2 systems for data centers and a general idea of the system was starting to be developed. The document explains a system in which air is conducted through the servers until a heat exchanger with CO_2 as working fluid. This was very useful in order create an initial concept of the desired system, even though not definitive. This idea is based on a bottom cooling system, this means, the heat exchangers where directly placed in the back part of each server rack, which is away from the idea of designing a modular system.

Having this on mind, the search kept on and another interesting idea was found in [21] for the modular concept to be developed. Google has developed a system in which a the hot isle of the data center was transformed in an unique structure which includes the heat exchangers to capture the waste heat and the fans to move the cooling air, but the coolant used in this case was water. That is why, by putting together this two technologies, the main idea of the project was already set.

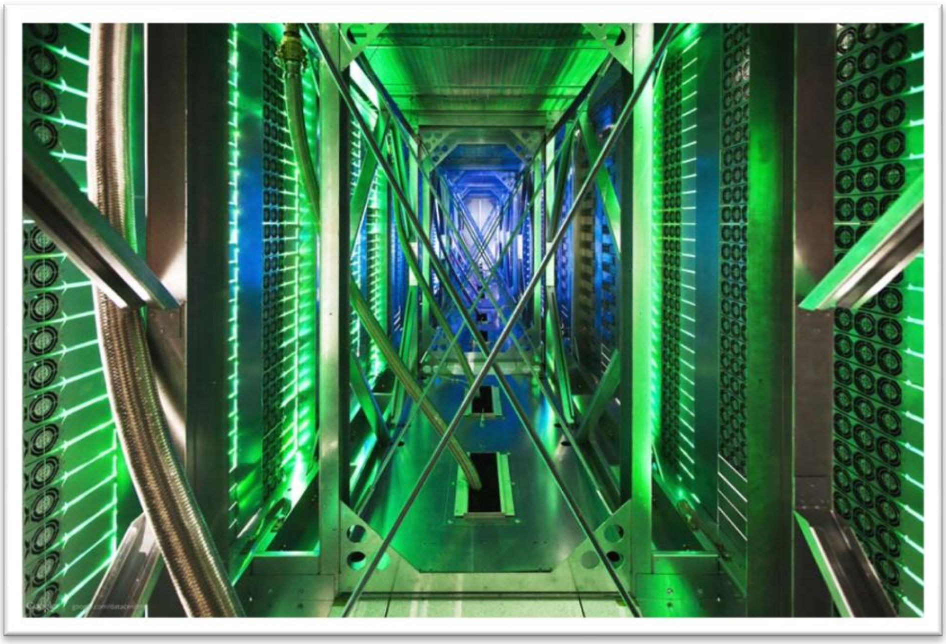

Figure 3.1. View of the hot isle of a Google data center (Source: [21])

After this, further information was needed in order to develop a further design of the systems. Thus, the next step followed consisted in searching in the web of the European Patent Office (EPO) called EPACENET, in order to find the deeper documentation about the ideas. Both the TROX AITCS [22] and the Google data center [23] were found, what allowed to develop initial designs and schedules of the modular system.

This documents, own ideas and further information checked about modular way of designing data centers helped to finally create the actual system.

At this point, the three different types of modules were created, after different measurements evaluations:

- **IT module:** This modules will contain all the racks with the servers and the power units. Furthermore, it will have all the cables and secondary installations needed and space for moving inside.
- **Cooling module:** The heat exchangers for the IT equipment and the fans for conducting the air will be placed in this module.
- **Heat management module:** All the equipment like pumps, condensers and evaporators will be placed inside. There will be three kinds of heat management module: one for heat rejection (a thermosyphon loop) and two for use of the waste heat (one with ammonia for space heating and other with CO_2 for water heating).

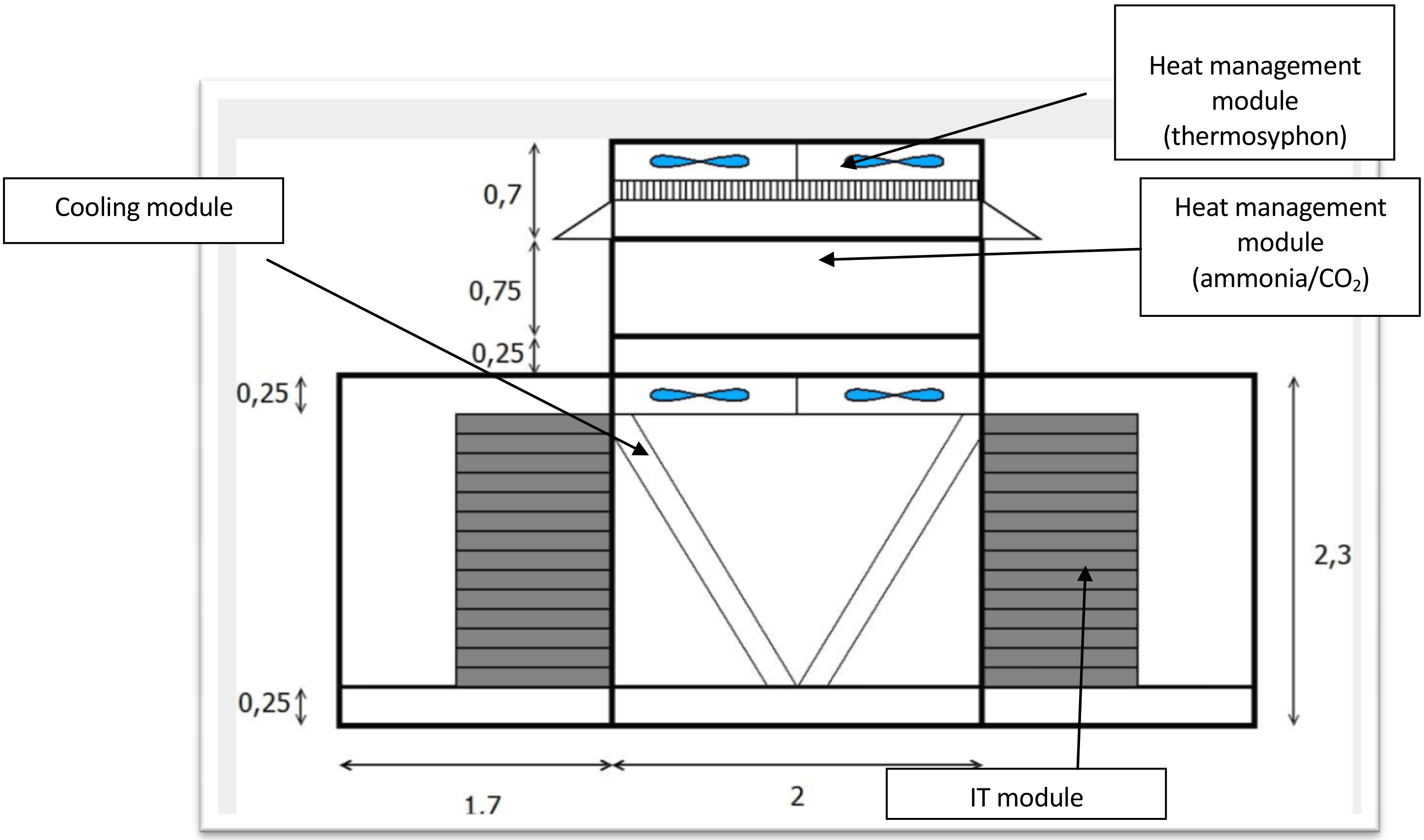

Figure 3.2. Distribution and measures of the data center (Source: Own design)

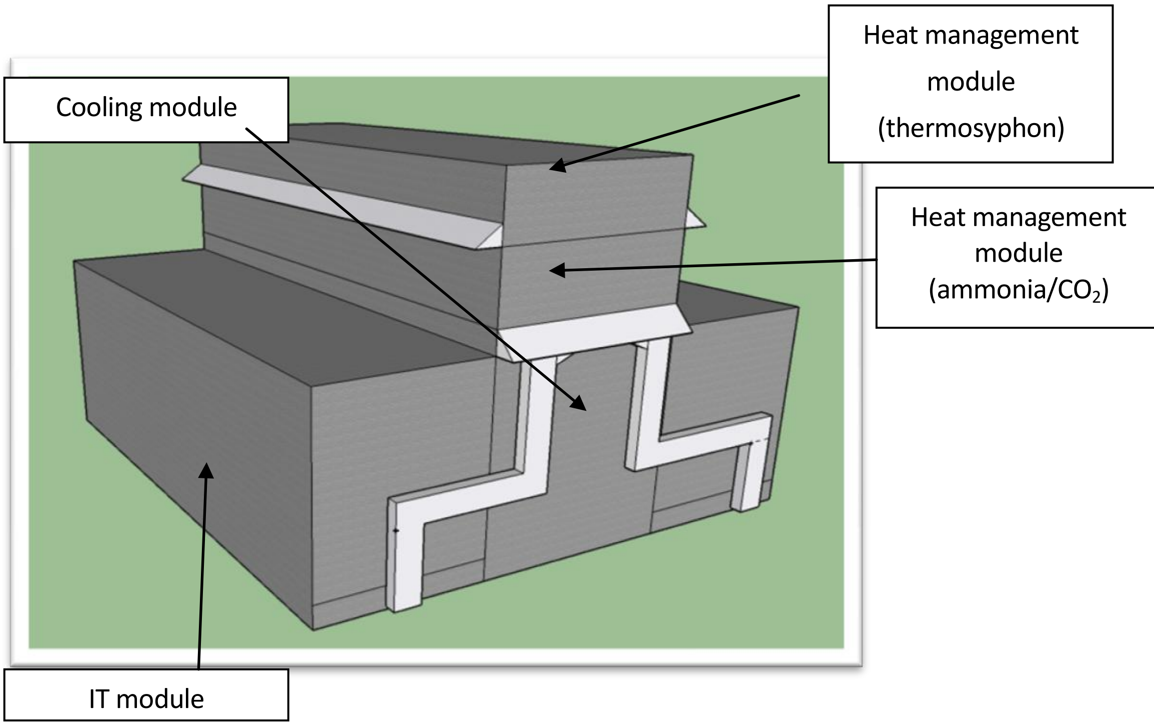

Figure 3.3. 3D scheme of the data center (Source: Own design)

Figures 3.2 and 3.3 show the distribution of the modules that create the whole structure. The IT modules will have the 12 server racks placed in a way that they have the bottom part looking to the central cooling module, and there will be no wall neither in the IT module nor in the cooling one in the part they are connected.

Additionally, the waste heat management modules will be placed on top of the cooling one. In the thermosyphon configuration, only the thermosyphon is set in the system to directly reject the heat. In the other two configurations (the ammonia-CO_2 and CO_2-CO_2), the correspondent module is set on top of the cooling one and the thermosyphon is placed on top of it just in case heat rejection is needed.

3.2. IT module

First of all, the different parameters that concern to the IT module have to be decided. The measures of a standard server rack were checked in order to know how many of them is possible to adapt in a container. As reference, the approximate measures of a 19' rack were taken into account for the calculations. The standard length for containers are 20 feet and 40 feet, and in here the first type was selected. Furthermore, the amount of heat produced is shown, with the other measures, in the table below.

Height [m]	1,8
Length [m]	1
Width [m]	0,5
Heat load [W]	26000

Table 3.1. Server rack data

Some initial measures for the IT container were selected in order to be able to do a first approach of the size of the heat exchanger, so modifications on the size were done afterwards in order to find a middle point between both the container and the heat exchanger.

Apart from the length, that is already a standard measure (either 20 or 40 feet; the first one this case), both the width and the height were decided by reviewing different container fabricants and also, taking into account the rack measures, by estimating the space needed for other equipment as cables, tubes... and obviously for persons to move inside. For example, as it can be seen in [24] this company uses containers of 3,55 meters width, but the figure below shows two corridors have been used, one on each side of the servers, which doesn't correspond with the study case.

Figure 3.4. Inner view of a data center (Source: [24])

Furthermore, the racks have 1 meter depth so, taking into account how much a person need to walk through the container, the size was decided. The table below shows the values assigned.

IT container measures (m)	
Length	6,1
Width	1,7
Height	2,3

Table 3.2. IT container measures

In order to now the total amount of heat produced by, the calculation is very simple.

$$\dot{Q}_{total} = \frac{Module\ length}{Rack\ width} \cdot \dot{Q}_{rack} \qquad \textbf{(Eq. 3.1)}$$

The racks inside the container will be placed in a way that their back part will work as the wall of the container, as this will be open in one of the sides. This is due to the fact that the container will be directly connected to the cooling one, and it needs free opening in order to let the air flow through the heat exchanger.

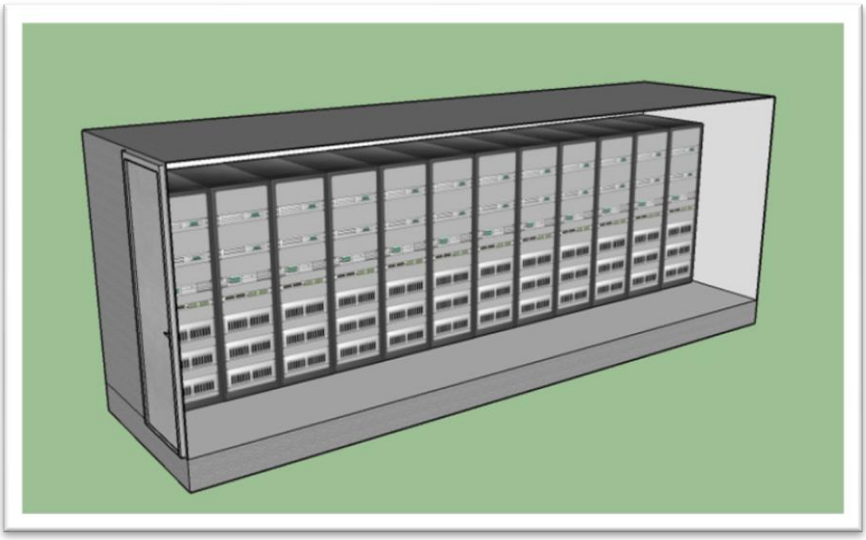

Figure 3.5. 3D model of the front part of the IT module (Source: Own design)

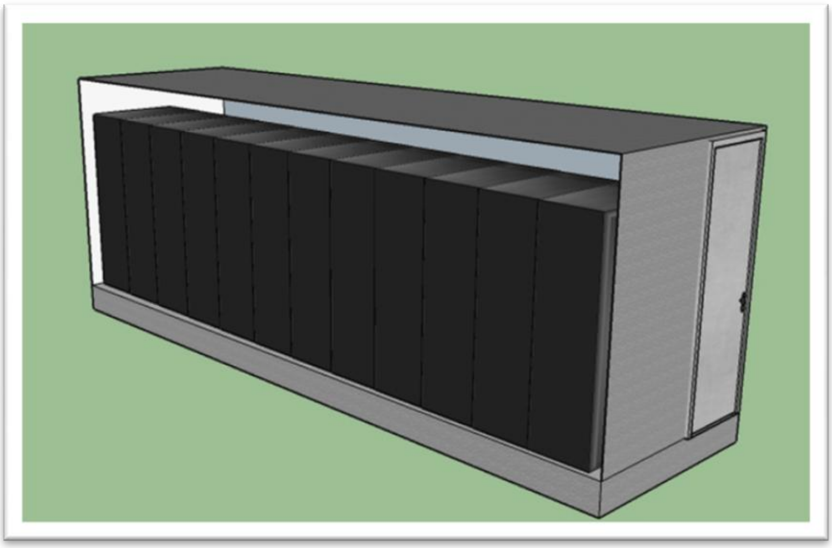

Figure 3.6. 3D model of the back part of the IT module (Source: Own design)

In figures 3.5 and 3.6, it is shown the configuration decided for the IT module. The sketches are not exactly how the system would actually be, but their purpose is to get an idea of how the racks would be placed in the container.

The figures below show very rough schemes of the different points of view of this module, in order to understand the measures distribution.

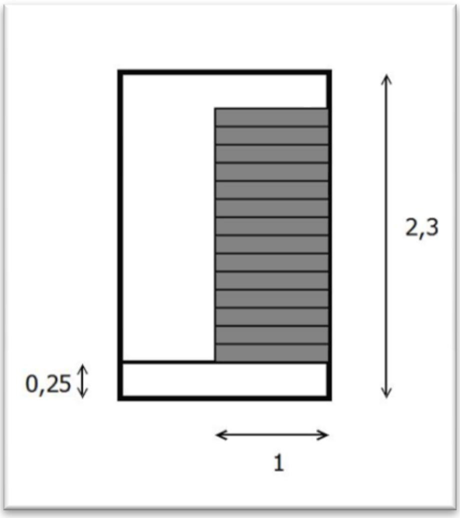

Figure 3.7. Front view and measures of the data center (Source: Own design)

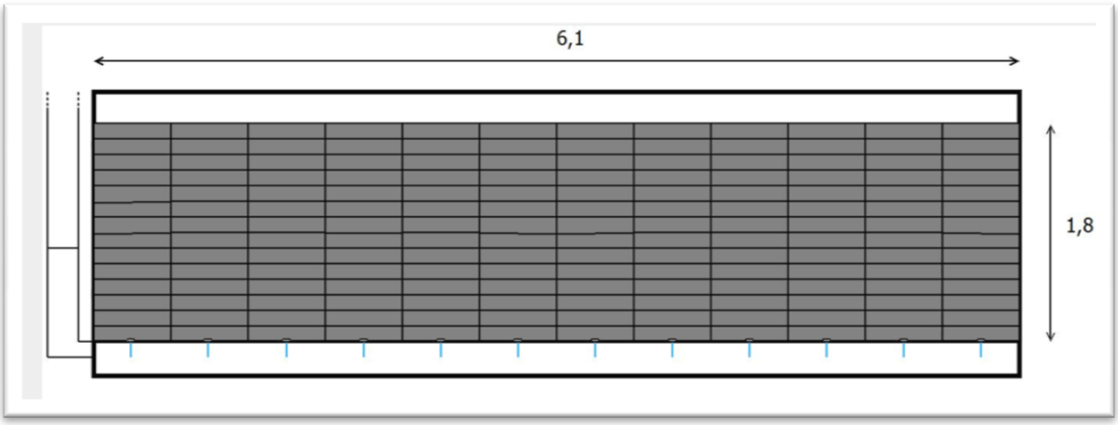

Figure 3.8. Side view and measures of the data center (Source: Own design)

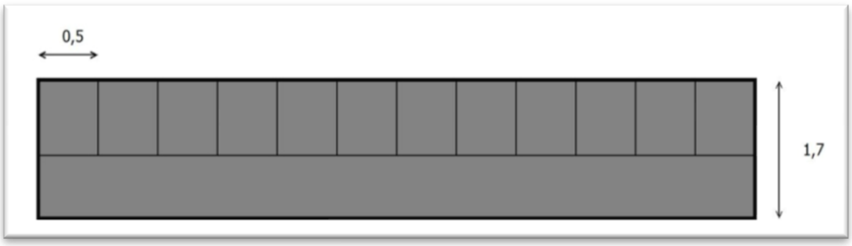

Figure 3.9. Top view and measures of the data center (Source: Own design)

3.3. Cooling module

After the IT module is sized, the cooling one has to be too, always respecting the previous measures, adapting to the needs and following the existing standards. The length is already decided as it has to be the same of the prior case. The width was determined following, first of all the idea of the cooling module of [23]. The heat exchangers will need space as they will have an angle (if they are totally straight, heat transfer does not occur properly), but also not totally horizontal, as then the heat exchanger will be way too big and thick, unless a very big container is designed, which is not desired. Thus, taking into account that the height of the server racks is 1,8 meters, the width shown in table 3.3 was selected. Finally, based also in the server rack height, the one of the cooling module was selected, taking into account extra space that will be needed for other devices.

Cooling container measures (m)	
Length	6,1
Width	2
Height	2,55

Table 3.3. Cooling container measures

The cooling module will be placed in the middle of two IT modules, so this leads to a design of the heat exchanger that adapts to this need properly, without taking too much space. The picture shows how the device will be fixed, forming an angle so its useful height can be maximized and below it, pictures 3.11 and 3.12 show an scheme with the measures.

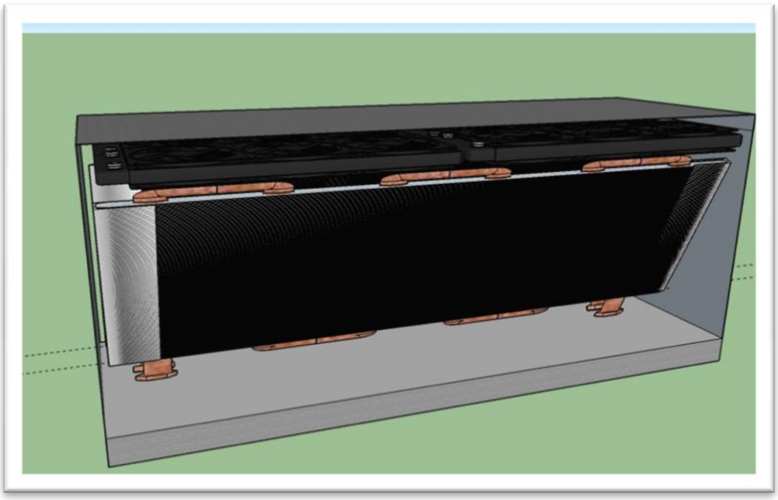

Figure 3.10. 3D model of the cooling module (Source: Own design)

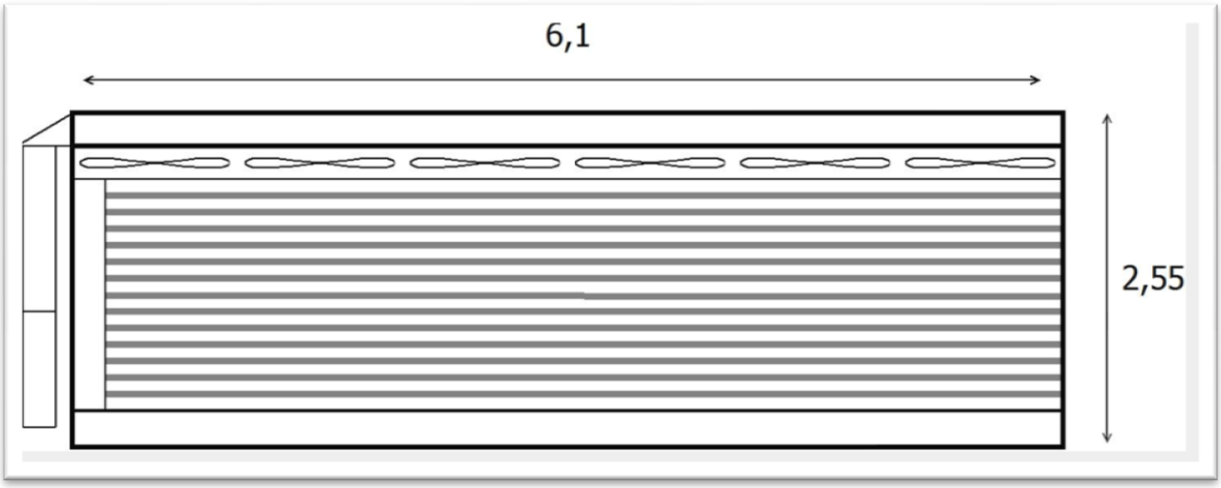

Figure 3.11. Side view of the cooling module and measures (Source: Own design)

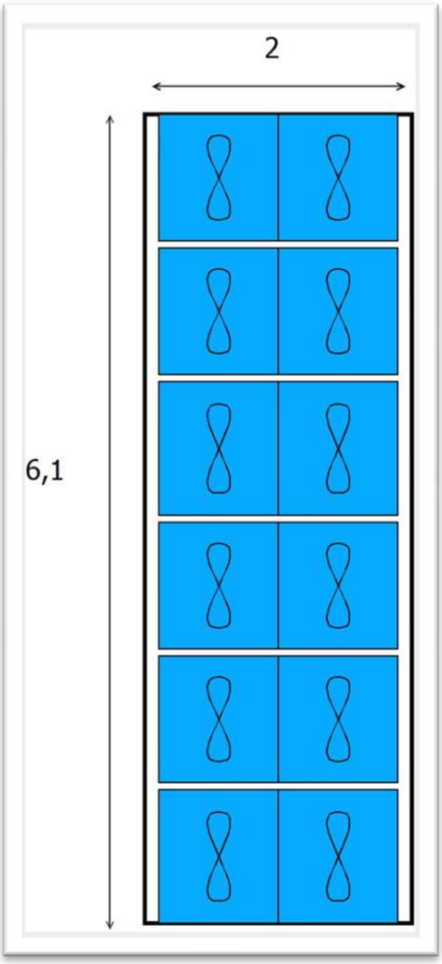

Figure 3.12. Top view of the cooling module and measures (Source: Own design)

For calculating the useful height of the heat exchanger the formula 3.2 was used, in which the height (h) and half of the width (w) of the cooling module are involved.

$$H = \sqrt{h^2 + \left(\frac{w}{2}\right)^2} \qquad \textbf{(Eq. 3.2)}$$

The next steps in the design will be sizing the heat exchanger and calculating the fan power needed to move the air through the servers.

3.3.1. Heat exchanger sizing

As the heat load the cooling module has to put up with can be assumed as a constant, it is possible to design a heat exchanger that suits that need. The formula that is going to be used for this task is 3.3 and the process followed is the one shown in [25].

$$A_{exchange} = \frac{Q}{U \cdot \Delta Tm} \qquad \textbf{(Eq. 3.3)}$$

A table summarizing all the measures involved in the heat exchanger design that are going to be used in this section is provided below.

Description	Symbol
Fin effectiveness [-]	η
Convection heat transfer coefficient of air [W/m^2·K]	h_{air}
Convection heat transfer coefficient of the refrigerant [W/m^2·K]	h_{ref}
Fin-to-tube ratio [-]	B
Nusselt number [-]	Nu
Thermal conductivity [W/K·m]	λ
Reynolds number [-]	Re
Prandtl number[-]	Pr
Density of the fluid [kg/m^3]	ρ
Velocity of the fluid [m/s]	C
Dynamic viscosity [kg/m·s]	μ
Mass flow [kg/s]	$\dot{m}$
Specific heat capacity of the air [J/kg·K]	Cp
Temperature difference [ºC]	ΔT
Height of the heat exchanger [m]	h_{HX}
Length of the heat exchanger [m]	l_{HX}
Number of tubes per layer [-]	$n_{tubes-layer}$
Number of layers [-]	n_{layers}
Tube diameter [m]	d_o
Number of fins per tube [-]	$n_{fin-tube}$
Thick of the fins [m]	δ
Separation between fins [m]	a
Width of the fin per tube [m]	l_h
Height of the fin per tube [m]	l_v
Area of the finned tube [m^2]	A
Area of the bared tube [m^2]	A_{bc}
Area that air crosses [m^2]	A_c

Table 3.4. Description and symbol of all the parameters in this section

First of all, it is necessary to define the temperatures of the different points in order to be able to calculate the ΔTm. This value is the logarithmic difference of temperature and follows the formula x.

$$\Delta Tm = \frac{\Delta T1 - \Delta T2}{\ln\left(\frac{\Delta T1}{\Delta T2}\right)} \qquad \textbf{(Eq. 3.4)}$$

Where, taking into account that counter flow heat exchange is supposed:

$$\Delta T1 = (T_{in,hot} - T_{out,cold}) \qquad \textbf{(Eq. 3.5)}$$

$$\Delta T2 = (T_{out,hot} - T_{in,cold}) \qquad \textbf{(Eq. 3.6)}$$

In this situation, the air is working as hot fluid while the CO_2 is the cold one. It must be stand out that, as the CO_2 is not suffering a change of state (latent heat is being used) its temperature will be constant.

Later, the overall heat transfer coefficient U must be calculated. This calculations is a bit more tedious than the one made for the temperature, so it must be followed step by step.

$$U = \frac{1}{\frac{1}{\eta \cdot h_{air}} + \frac{B}{h_{ref}}} \qquad \textbf{(Eq. 3.7)}$$

First of all, the h factor of the refrigerant must be calculated. In this case, the literature was reviewed in order to obtain this value which, for the temperature of 20 ºC, is **4242 W/m²·K**. From the same source, the Fin-to-tube ratio was extracted, with a value that can be between 10 and 30 for finned coils. A ratio of **10** was chosen.

The convection coefficient of air was calculated following the method shown in 3.8. First of all, the formula that gives the value of this coefficient is the one above.

$$h_{air} = \frac{Nu \cdot \lambda}{d_o} \qquad \textbf{(Eq. 3.8)}$$

Both the thermal conductivity and the diameter are values already settled, but in order to obtain the adimensional Nusselt number, either the formula 3.9 or the 3.10 must be used. The first one is referred to an inline arrangement of the tubes, while the second one is used for staggered arrangements.

$$Nu_{inline} = 0{,}22 \cdot Re^{0{,}6} \cdot \left(\frac{A}{A_{bc}}\right)^{-0{,}15} \cdot Pr^{\frac{1}{3}} \qquad \textbf{(Eq. 3.9)}$$

$$Nu_{staggered} = 0,38 \cdot Re^{0,6} \cdot \left(\frac{A}{A_{bc}}\right)^{-0,15} \cdot Pr^{\frac{1}{3}}$$ **(Eq. 3.10)**

The already well known formula of the Reynolds was used, then, in this calculation.

$$Re = \frac{\rho \cdot c \cdot d_o}{\mu}$$ **(Eq. 3.11)**

For calculating the velocity of the air passing through the heat exchanger the following process was carried on.

First of all, the air mass flow was calculated based on the temperatures and the heat produced by the IT container.

$$\dot{Q} = \dot{m} \cdot cp \cdot \Delta T$$ **(Eq. 3.12)**

This mass flow allows to calculate the velocity of the air, first of all applying the value of the air density and, afterwards, dividing by the area that the air passes through ($\boldsymbol{A_c}$).

$$c = \frac{\dot{m}}{\rho \cdot A_c}$$ **(Eq. 3.13)**

The figure below helps to understand which is the area that needs to be calculated in order to being able to know the velocity.

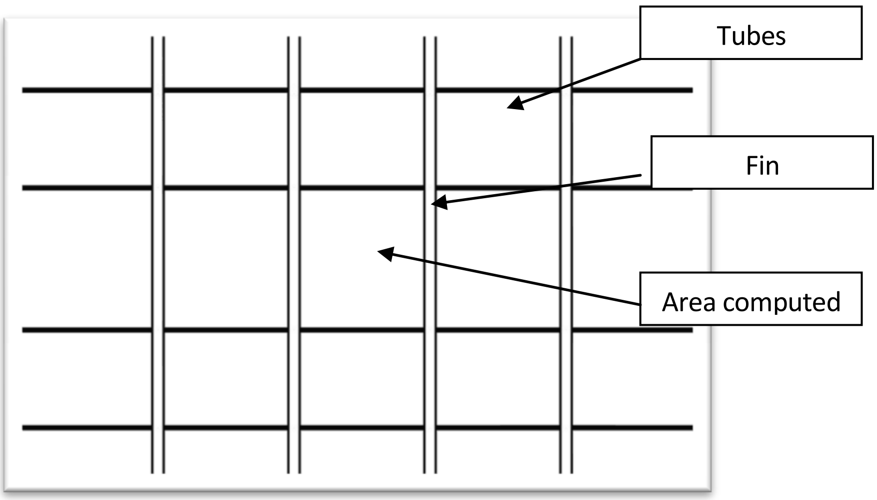

Figure 3.13. Scheme of the area used for the velocity calculation(Source: Own design)

So the formula following formulas were applied for calculating the area in staggered and linear configuration respectively.

$$A_{c,staggerd} = (lf - D) \cdot a \cdot \left(n_{fin} - 1\right) \cdot (n_{tubes-layer} - 1) \qquad \textbf{(Eq. 3.14)}$$

Once the velocity is known, the Reynolds number can be calculated and it will be used for calculating the Nusselt one, which will finally allow to obtain the value of the heat transfer coefficient for the air. However, before that, the area of the bared tube (A_{bc}) and the total area of the tube (A) have to be calculated.

The first one will be simply calculated with the formula 3.15.

$$A_{bc} = l \cdot \pi \cdot \mathrm{d}_0 \qquad \textbf{(Eq. 3.15)}$$

Unlike, calculating the total area requires a longer process for obtaining its value. First of all, the area of a single fin has to be calculated. For that, as the arrangement of the tubes is staggered, it is necessary to define a differential of area for each tube, which will be an hexagon, so the fin area of the tube can be known. Figure 3.14 shows how this differential is.

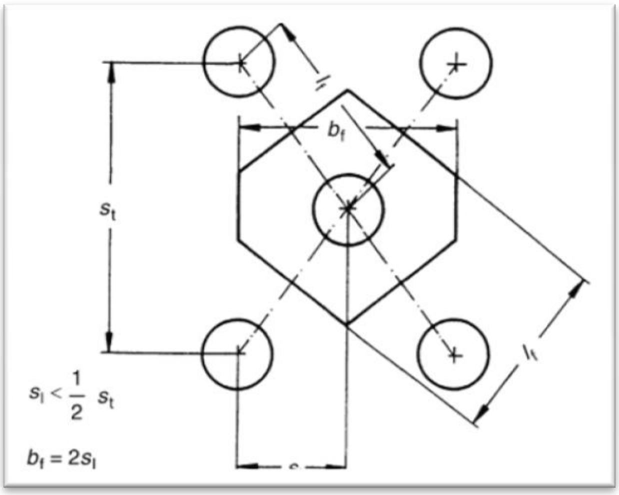

Figure 3.14. Tubes a fin view (Source: [25])

In the figure 3.15 the angles needed to compute the area of the fin can be seen. The short side of the hexagon has d_0 length, and the long side, a, has to be calculated by means of these angles. Firstly, then, the angle alpha is calculated for being able to calculate beta.

$$\alpha = arctag\frac{\frac{d_0}{2}}{\frac{b_f}{2}} \tag{Eq. 3.16}$$

$$\beta = 2 \cdot \alpha \tag{Eq. 3.17}$$

The fact of knowing beta is going to allow to calculate gamma, which will be divided in order to finally obtain the side b.

$$\gamma = \frac{360 - 2 \cdot \beta}{2} \tag{Eq. 3.18}$$

$$\epsilon = \frac{\gamma}{2} \tag{Eq. 3.19}$$

$$\tau = \frac{\epsilon}{2} \tag{Eq. 3.20}$$

$$b = tag\,\tau \cdot l_f \tag{Eq. 3.21}$$

Now, the area of the fin can be calculated, by means of the formula 3.22.

$$A_{fin} = \frac{b \cdot \frac{l_f}{2}}{2} \cdot 4 + d_0 \cdot \frac{b_f}{2} \tag{Eq. 3.22}$$

Finally, the total area will have two terms. The one previously computed of the fin area and the one that involves the area of the tube, leading to the formula 3.23.

$$A = A_{fin} \cdot n_{fin} + \pi \cdot d_0 \cdot a \cdot (n_{fin} + 1) \tag{Eq. 3.23}$$

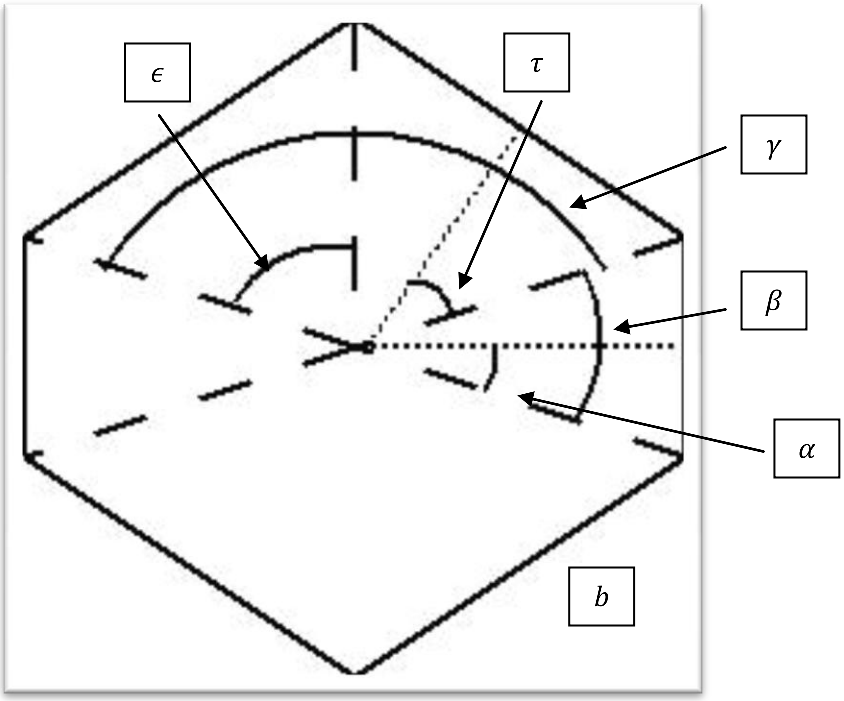

Figure 3.15. View of the hexagonal fin due to staggered configuration (Source: Own design)

The last value that needs to be calculated for the formula is the fin effectiveness. The process followed for obtaining that value is shown in formulas from 3.25 to 3.27 in the case of staggered configuration. Only formula 3.28 is different in the case of linear configuration.

$$\eta = \frac{\tanh X}{X} \qquad \textbf{(Eq. 3.24)}$$

$$X = \mathrm{\phi} \cdot \frac{\mathrm{d_0}}{2} \cdot \sqrt{\frac{2 \cdot \mathrm{h_{air}}}{\mathrm{\lambda_f} \cdot \mathrm{\delta}}} \qquad \textbf{(Eq. 3.25)}$$

$$\phi = (\phi' - 1) \cdot (1 + 0{,}35 \cdot \ln \phi') \qquad \textbf{(Eq. 3.26)}$$

$$\phi' = 1{,}27 \cdot \frac{b_f}{d_0} \cdot \sqrt{\frac{\mathrm{l_f}}{\mathrm{b_f}} - 0{,}3} \qquad \textbf{(Eq. 3.27)}$$

$$\phi' = 1{,}28 \cdot \frac{b_f}{d_0} \cdot \sqrt{\frac{\mathrm{l_f}}{\mathrm{b_f}} - 0{,}2} \qquad \textbf{(Eq. 3.28)}$$

3.3.2. Fan airflow

For being able to know the fan power, the air volumetric flow has to be calculated. As the inlet and outlet temperatures are already set, only the cp and afterwards the density of the air in those conditions are needed and, by means of the formula 3.29, the flow will be obtained.

$$\dot{V} = \frac{Q}{cp \cdot \Delta T \cdot \rho_{air}} \qquad \textbf{(Eq. 3.29)}$$

3.4. Thermosyphon loop module (Configuration 1)

This module is the first of the three ones that are designed for making use of the heat captured by the cooling module. In this case, the goal of the module will be only rejecting the heat to the outside. It is thought in this way as in some situations, depending on the outside conditions, heat recovery for whatever usage might not be necessary, so this kind of circuit that doesn't need electric power from a pump is desirable.

The thermosyphon loop bases its heat transfer method in natural convection. The driven force, then, is created due to a difference of density in the fluid. As the heat is absorbed and the temperature increases, the fluid becomes less dense which leads to a gradient in this property that creates the flow in the circuit. As the coolant passes through the evaporator, it becomes vapour which makes that it rises upstream the pipes until the condenser, where the heat is released so the temperature goes down and the fluid descend through the cold pipe.

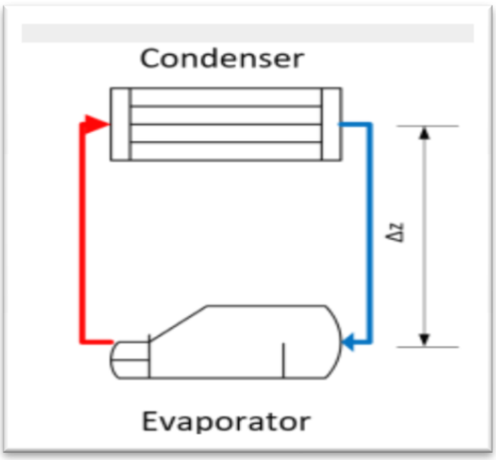

Figure 3.16. Thermosyphon loop scheme (Source: [26])

The formula that expresses the variation in the pressure in this system is 3.30 extracted from[26], which also allows to calculate the forced created by the fluid if interested.

$$\frac{\overrightarrow{F_A}}{A} = \Delta p = (\rho_l - \rho_v) \cdot \vec{g} \cdot \Delta z \qquad \textbf{(Eq. 3.30)}$$

Regarding the structure of this module, it will be formed exclusively by a condenser and a series of fans that will create the air circulation necessary to cool down the CO_2 in it. Apart from that, it will have pipe connections for linking it with the cooling module and air entries.

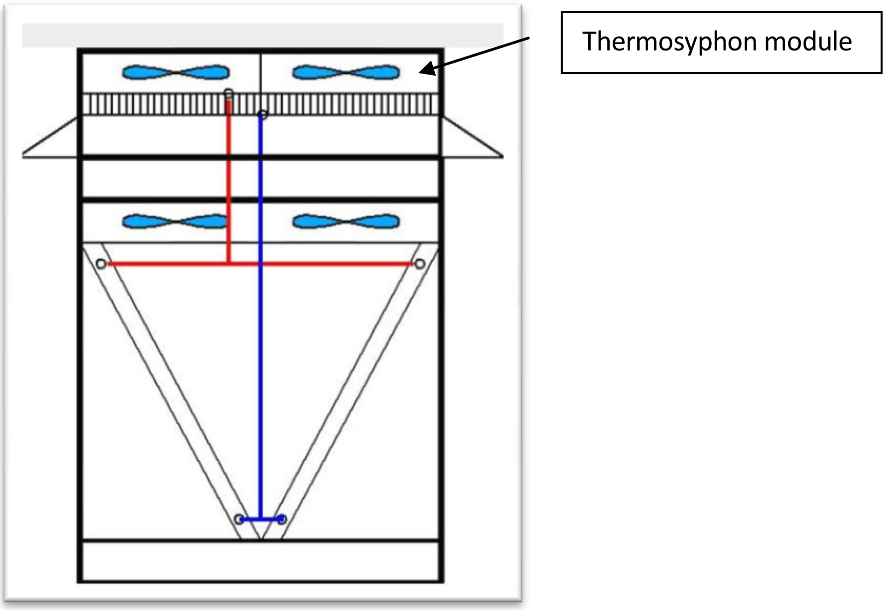

Figure 3.17. Scheme of the front view of the thermosyphon loop CO_2 pipe connection (Source: Own design)

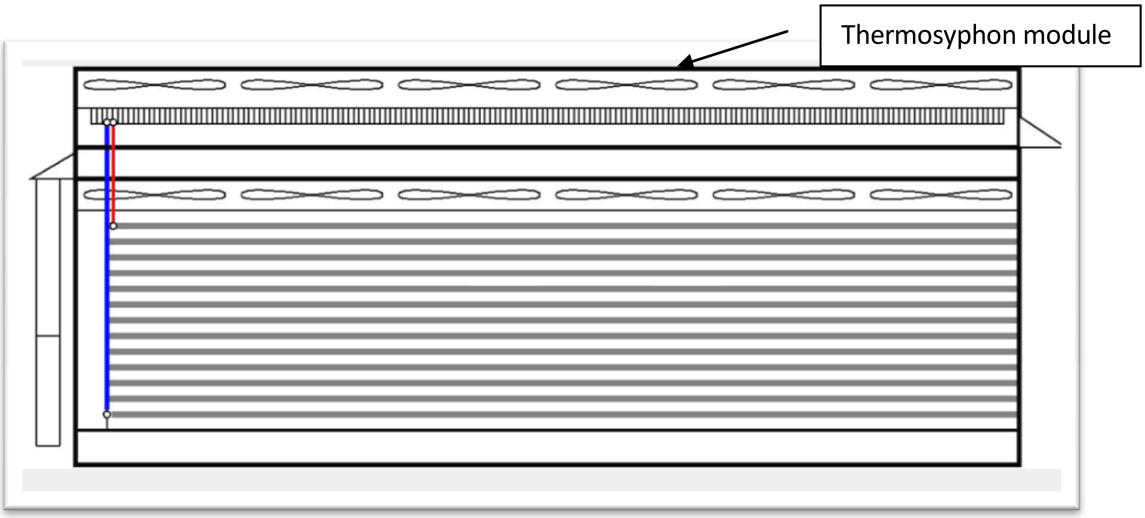

Figure 3.18. Scheme of the side view of the thermosyphon loop CO_2 pipe connection (Source: Own design)

Figures 3.17 and 3.18 show roughly how it will be the shape of the cooling and the thermosyphon module together. The CO_2 goes down the cold pipe (blue one) and enters the two evaporators in the cooling module, where the heat transfer is produced, and it rises through it until the outlet which is the hot pipe (red one). Following this tube, the coolant arrives to the condenser where air is circulating thanks to a series of fan and it extracts the heat from it rejecting it to the outside.

The table below shows the values that are set in system. The heat will be absorbed by using the change of state of the CO_2 so the temperature will be constant.

CO_2 parameters	
Temperature	20 ºC
Pressure	57,29 bar
Saturated liquid enthalpy	255,8 kJ/kg
Saturated vapour enthalpy	407,9 kJ/kg
Enthalpy difference	152,1 kJ/kg

Table 3.5. Initial CO_2 data for the thermosyphon calculations

Formula 3.31 is a variation of 3.12, and it is used in cases where the coolant that captures the heat is working exclusively with change of state to calculate the mass flow through the pipes.

$$\dot{m} = \frac{\dot{Q}_{total}}{\Delta h_{latent\ heat}} \quad \textbf{(Eq. 3.31)}$$

As it was said before, this circuit is pumpless so it has to be designed, regarding the pipe section, in a way that the pressure drop calculated with the Darcy-Weisbach equation doesn't overcome the one computed with the formula 3.29, as this will be the limit. Formulas from 3.31 to 3.33 are the ones used for making this calculation.

$$h_f = 0{,}0826 \cdot f \cdot \frac{Q^2}{D^5} \cdot l \quad \textbf{(Eq. 3.32)}$$

$$f = \frac{0{,}25}{\left(\log\left[\frac{\epsilon}{3{,}7 \cdot D} + \frac{2{,}51}{Re \cdot \sqrt{f}}\right]\right)^2} \quad \textbf{(Eq. 3.33)}$$

$$\epsilon_r = \frac{\epsilon}{D} \quad \textbf{(Eq. 3.34)}$$

It is important to stand out how the length used in formula 3.32 is calculated. Not only the length of the pipes is taken into account, but also the equivalent length of the different elements in the circuit, as valves, corners, etc.

$$l_{total} = l_{pipe} + l_{elements} \qquad \textbf{(Eq. 3.35)}$$

Also, when obtaining the pressure drop of the heat exchangers with the HXSim simulations, equation 3.35 has to be taken into account in order to convert the pressure in equivalent length.

$$l_{equivalent} = \frac{\Delta p}{\rho_{coolant}} \qquad \textbf{(Eq. 3.36)}$$

A Matlab code that can be checked in Annex A1 was designed in order to be able to calculate the diameter allowed in a quick way.

3.5. CO_2-Ammonia circuit module (Configuration 2)

3.5.1. Design

As in the other cases, initially the measures of the module are shown in table 3.6. In this case, its measures were limited first off all by the total height of the system that is set that will not overcome 4 meters. However, there has to be space enough for all the equipment and the module has to adapt to the cooling one, as it will be placed on top of it.

CO_2-Ammonia module measures	
Length	6,1 m
Width	2 m
Height	0,75 m

Table 3.6. Module measures

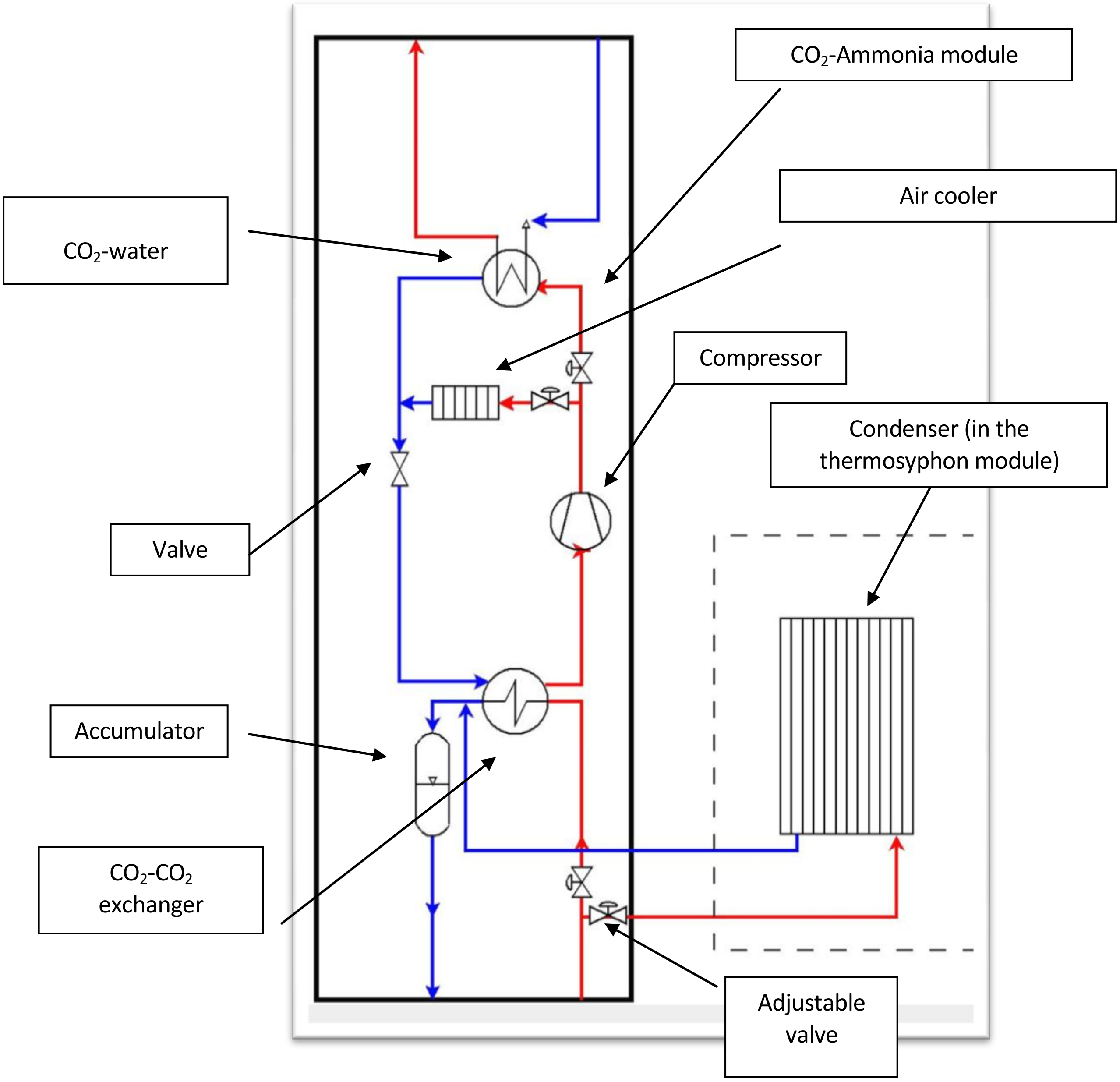

Figure 3.19. Scheme of the CO_2-Ammonia circuit (Source: Own design)

Figure 3.19 shows how it is the design of the module, which will be formed by two circuits. The first one is the CO_2 one that captures the heat from the data center, so it is connected to the evaporator in the cooling module. Its elements will be a condenser that releases the heat in the ammonia circuit, an accumulator and adjustable valves.. The second loop is the NH_3 one, where all the heat extracted from the CO_2 circuit is transmitted to an ammonia heat pump that, by means of a compressor, rises the pressure and the temperature level of the circuit in order to transfer it to the water for space heating. Figure 3.20 and 3.21 show a rough idea of the physical appearance of the circuit, as well as its measures.

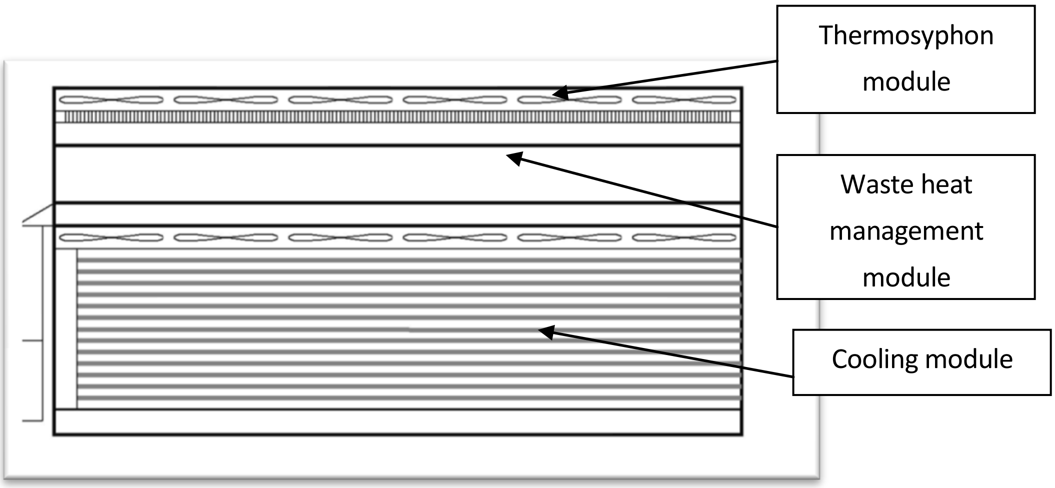

Figure 3.20. Scheme of the side view of the distribution of the modules (Source: Own design)

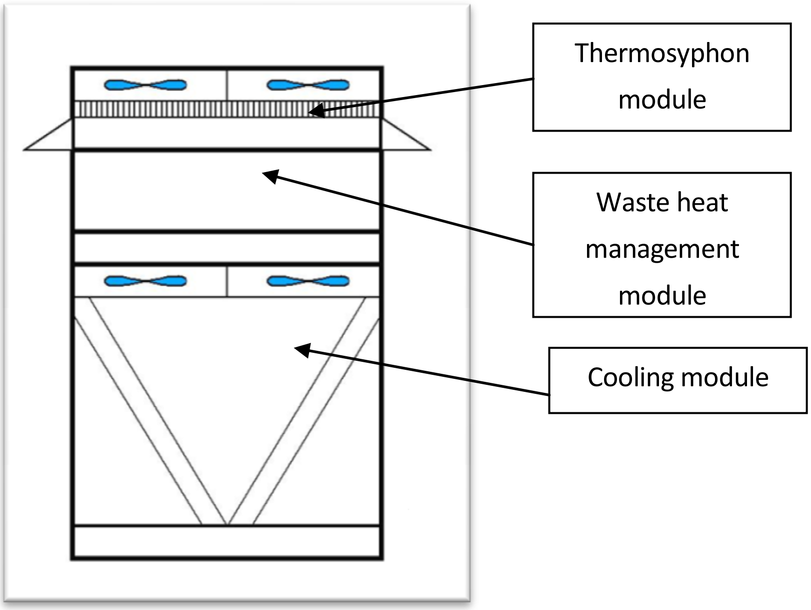

Figure 3.21. Scheme of thef ront view of the distribution of the modules (Source: Own design)

3.5.2. Operation modes

3.5.2.1. <u>Heat rejection (operation in winter)</u>

In this configuration, also the thermposyphon module has to be used. This is due to the fact that, even though this module is thought to reuse the waste heat, there might be moments when the heat is not needed so it has to be rejected to the environment. As the thermosyphon module uses air from the outside, it is important that the temperature of this air does not overcome the one that the CO_2 in the first loop has, 20 ºC. The cooler the outside air is, the better the performance of the cooling process is going to be, as the gradient of temperature regarding the CO_2 is bigger.

3.5.2.2. Heat rejection (operation in summer)

Problems can happen when using the thermosyphon loop to reject the heat to the outside when the temperatures are too high. Basically, the system will actually perform, but it will do it in a way that causes problems due to high temperature. The temperature in the thermosyphon condenser will be increasing from 20 ºC until higher values due to the high external temperature, which afterwards is translated in warmer CO_2 circulating in the loop. Consequently, also the temperature of the air that cools down the IT equipment will slowly but surely increasing, until it reaches a point that is not able to chill properly the servers and those start to malfunction.

This problem, however, has a simple solution. The heat from the CO_2 loop will be transferred to the ammonia one but, instead of releasing that heat to the water chiller, there will be a gas cooler connected in parallel, as shown in figure 3.19. As the ammonia temperature in the condenser is going to be high, even with elevated temperatures in the outside, is going to be possible to reject the heat properly.

3.5.2.3. Space heating

In spite of having the previous operation modes, the main objective of this module is to use the waste heat for space heating. As it is explained in the literature review, ammonia is one of the most common refrigerants nowadays as it shows very good performance due to its good thermal properties, which suits very well this kind of application.

In this mode, the heat absorbed by the CO_2 loop will be transferred to the ammonia one, to be transferred later to the water supply system and heat it up for space heating applications. Thus, the valve that drives the ammonia until the gas cooler will be closed, so the coolant follows the way until the condenser.

Once this values were fixed, the calculation of the different points of the thermodynamic cycle of ammonia was done using the software CoolPack, what allows to know the pressure levels and the enthalpy on each point of the cycle, and consequently to calculate the Coefficient of Performance of the heat pump (COP) by means of the formula 3.37. In this case, the COP calculated is the cooling one, as is the interesting for the purpose of the system.

$$COP_{cooling} = \frac{Q_{cooling}}{W_{pump/compressor}} = \frac{Q_{evaporator}}{W_{pump/compressor}} \quad \textbf{(Eq. 3.37)}$$

In this case, the height losses are negligible as the amount of pressure introduced by the pump is way bigger in relation that the one lost due to the friction in the tubes.

3.6. CO_2-CO_2 circuit module (Configuration 3)

3.6.1. Desgin

This is the third and last of the modules designed for reusing the waste heat produced by the data center. In this case, the main purpose is to use CO_2 as unique coolant of the system and using the amount heat for water heating, which is more demanding in terms of temperature than space heating.

This configuration will be formed by two loop system, both of them working with CO_2. The fact of having two separated loops is due to the effect of oils (needed for the pump to work) in the circuit. If only one loop was considered, the heat transfer from the hot to the cold side will not be produced under optimal conditions, as the oil has the effect of decreasing the quality of the heat transmission. Thus, even though adding an extra loop has also negative effects (as the general temperature level has to be decreased due to the intermediate heat exchange), this operation mode was selected as it ensures better performance.

The first of the circuits will be the have the same design as the first one in the ammonia module. Just the CO_2 accumulator, the condenser and the adjustable valves. The second CO_2 loop is formed by the evaporator that captures the heat from the first loop, the pump that increased the pressure level, condenser and reduction valve.

The physical appearance of the module will be the same as the one in the previous section, in pictures 3.20 and 3.21, and its size will adapt to the measures set in table 3.6.

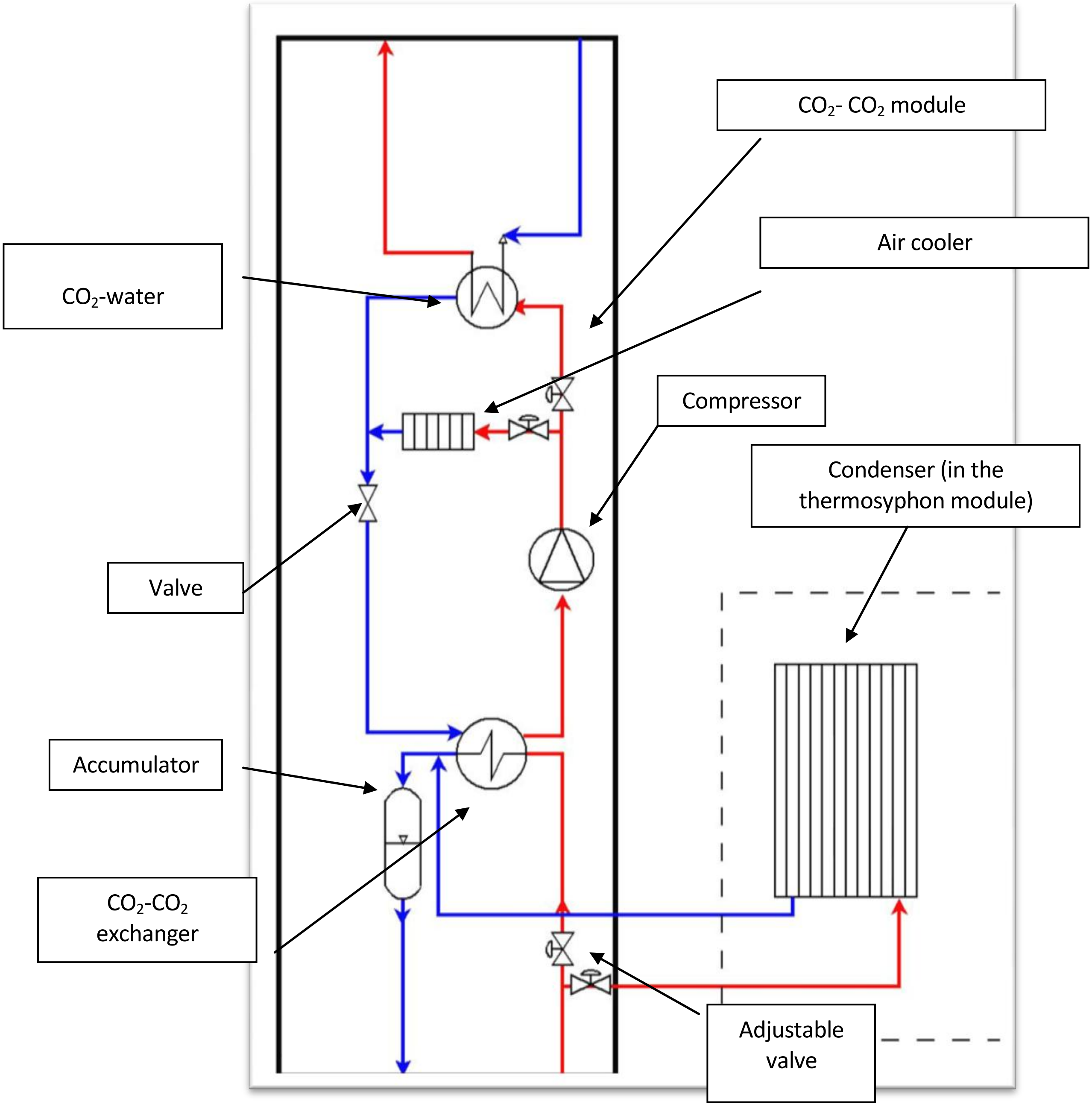

Figure 3.22. Scheme of CO_2-CO_2 circuit (Source: Own design)

3.6.2. Operation modes

The heat rejection operation modes of this module are the same than in the space heating module. The only difference that will appear is that, when rejecting the heat in summer operation, the CO_2 will have higher temperature than the ammonia in the condenser, so the performance will be a little better (best temperature gradient). For the rest, both system have the same performing.

3.6.2.1. Water heating

In this working mode, the operation will be the same as in the CO_2-Ammonia module, just with the difference that in here the reusing of heat will be done with CO_2 too. The other differences that will be is the actual use of that amount of energy. This configuration will work for heating the tap water from its lower temperature until a temperature of 90 ºC degrees that can be supplied to the dwellings.

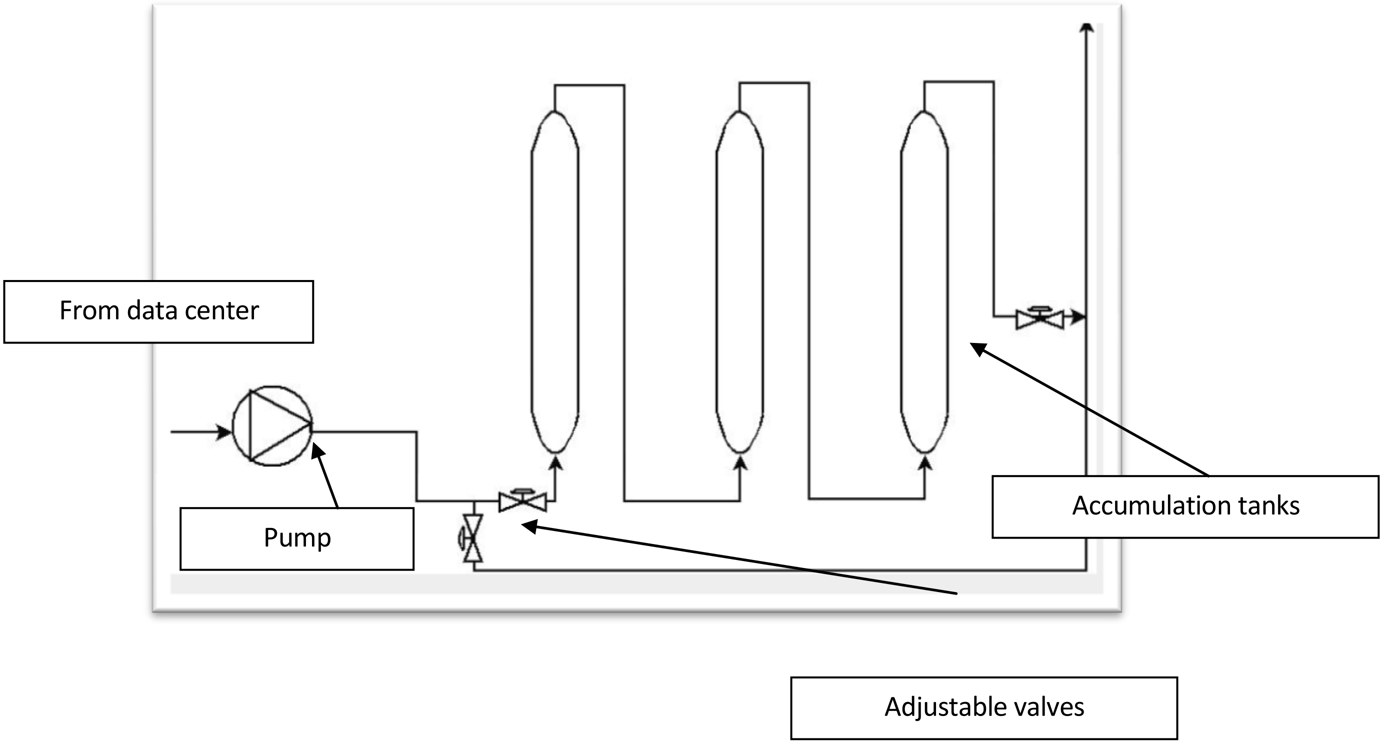

Figure 3.23. Scheme of the water supply system for water heating (Source: Own design)

Basically, the system will have several working modes. When the dwellings need warm water, the system will provide it either directly in the tanks are full, or through the tanks when they are empty. When there is no need of water heating in the households, the system will only operate if the tanks are not totally full. Once there is no more space for water, there will be a control system in the data center that will change from water heating mode to heat rejection mode.

3.7. Air circulation

3.7.1. Circulation through IT equipment

How is the air distribution and its movement through the different modules and parts of the data center is an important issue. It is shown in the following pictures.

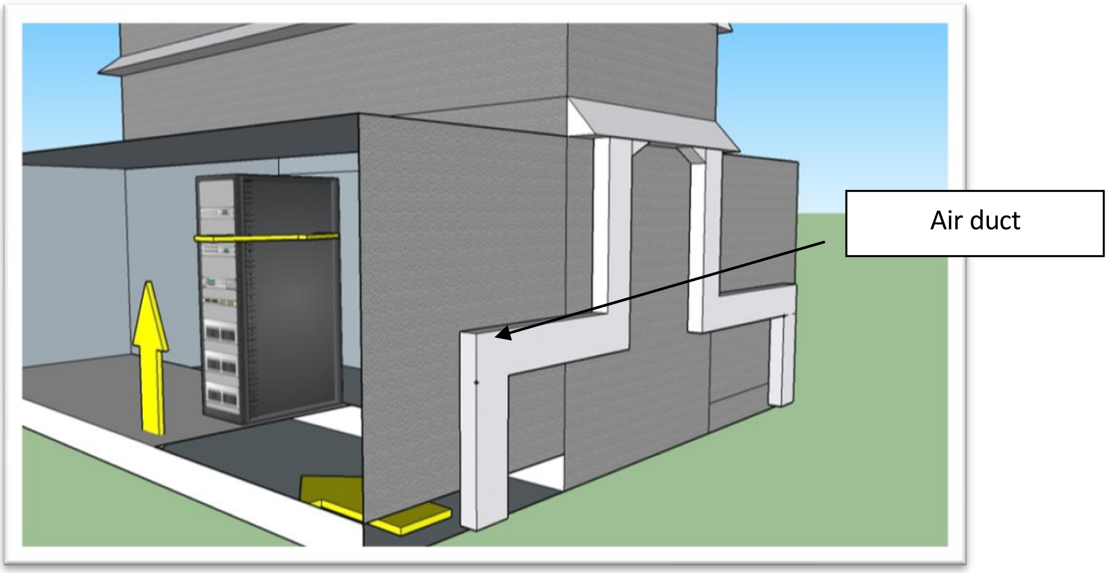

Figure 3.24. 3D model of the IT equipment air ciruclation (Source: Own design)

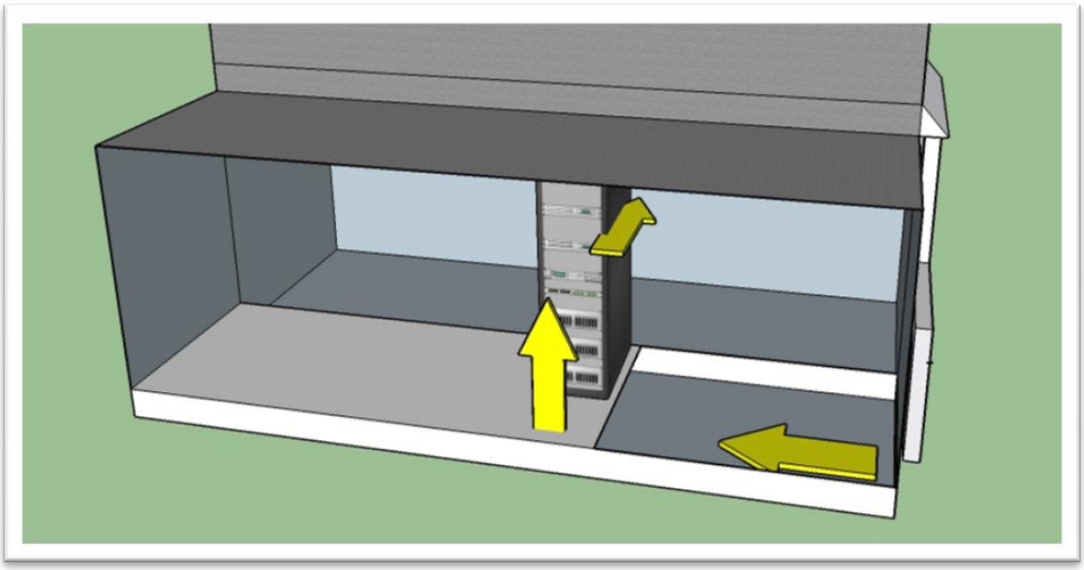

Figure 3.25. 3D model of the IT equipment air ciruclation (Source: Own design)

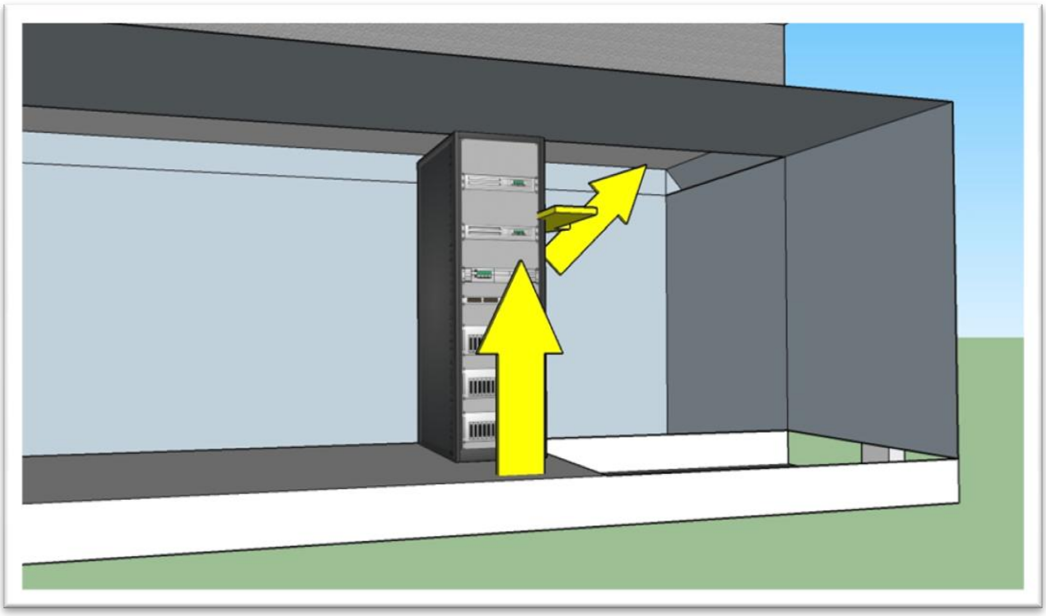

Figure 3.26. 3D model of the IT equipment air ciruclation (Source: Own design)

First of all, in figure 3.24 it can be seen the two connections between the cooling module and both of the IT modules. The air is pushed down through this tubes due to the suction power of the fans and it arrives until the servers module. Those servers will be placed on a pendum so the air will get under them and then raise up the through perforated floor, as it can be seen in figure 3.25. Afterwards, the air is forced to move through the IT equipment cooling it down until the desired temperature and arriving again to the cooling module, as figure 3.26 shows. Finally, the air keeps on rising through the fans until arriving to the empty volume on top of this module, where its conducted again through the connections with the IT modules.

3.7.2. Circulation in the thermosyphon loop

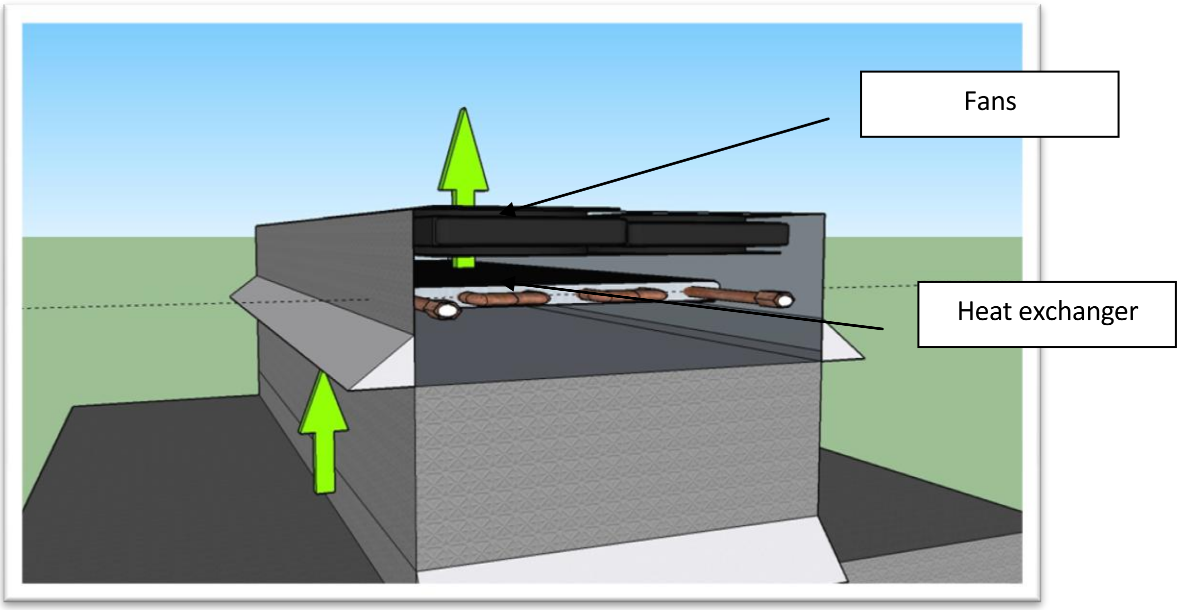

Figure 3.27. 3D model of the thermosyphon module air ciruclation (Source: Own design)

In this case, the air circulation is way more simple. There are openings on both sides of the thermosyphon module so the air is sucked thanks to the fans placed on top, and it passes through the condenser so the heat from the CO_2 is transferred to the air and rejected into the environment.

3.7.3. Circulation in the air cooler

The air circulation in this case is thought for the second operation mode explained in the module explanation sections, the one that operates when the temperature outside is too high to be able to cool the CO_2 in the thermosyphon. Figure 3.28 show the way the air will follow.

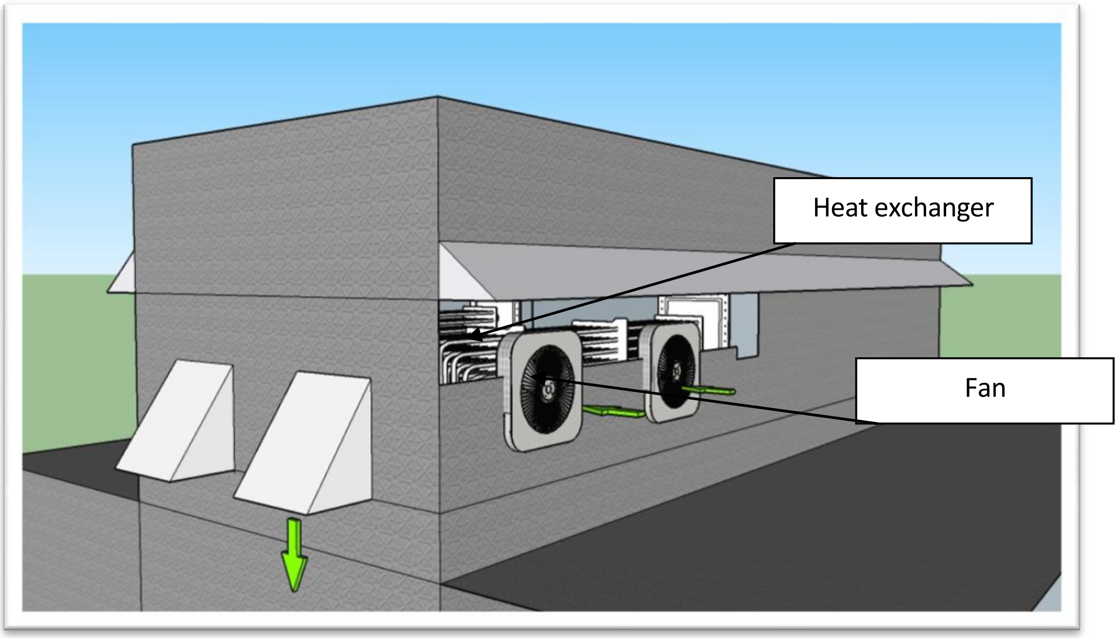

Figure 3.28. 3D model of air coolers module air ciruclation (Source: Own design)

As it can be seen, part of the waste heat management module will be divided to introduce there the air condensers needed, and an air entrance will be place on both sides of the module. Basically, the fans in the back part will create the lower pressure needed to make the air get into the entrance and reach the air cooler. There, either the ammonia or the CO_2 circuit will be connected to the condenser and the heat will be transferred to the air and then rejected to the outside.

3.8. Energy analysis

An energy analysis will be perform in order to evaluate the performance of the data center regarding energy savings and hot water supply the system is able to produce.

First of all, is important to know the concept of PUE (Power Usage Effectiveness) when talking about data centers. This concept talks about the amount of electric energy the data center has to use apart from the one the IT equipment is actually using, so it is important not know how good is the performance of the system. Thus, as explained in previous chapters, its minimum value has to be one. Formula 3.38 explains how it is.

$$PUE = \frac{Total\ energy\ consumption\ of\ the\ data\ center}{IT\ equipment\ consumption} \quad \textbf{(Eq. 3.38)}$$

Later, a small analysis the repercussion of the reuse of the rejected heat was done.

Basically, it is an evaluation of the amount of dwellings that can be supplied with the energy utilized. Thus, firstly, based on the data shown in table 3.7, formula 3.39 was used to calculate the total area that can be warmed up.

Total consumption per square meter [kWh/m^2] [27]	400
Space heating consumption [%] [27]	57
Water heating consumption [%] [27]	25
Average household area in Norway [m^2] [28]	115

Table 3.7. Initial data for energy analysis

$$A_{warmed} = \frac{\dot{Q}_{data\ center}}{C_{per\ area} \cdot SH} \qquad \textbf{(Eq. 3.39)}$$

Later, using the result of this calculation and knowing the area per house established in table 3.40, the number of dwellings was calculated.

$$Dwellings\ supplied = \frac{A_{warmed}}{A_{house}} \qquad \textbf{(Eq. 3.40)}$$

4. RESULTS

4.1. IT module

By means of the formula 3.1 it is possible to know how many server racks it is possible to place in one of the IT modules. This allows lately to know the amount of heat rejected by them which it is the starting point for further calculations. This way, table 4.1 shows the results obtained.

IT module length [m]	6,1
Rack length [m]	0,5
Number of racks per module [-]	12
Heat produced per module [kW]	312
Total heat produced [kW]	624

Table 4.1. Initial data the heat exchanger sizing

4.2. Cooling module

4.2.1. Heat exchanger sizing

The optimal sizing of the heat exchanger was determined by an iterative method between the own calculations made with Excel following the formulas in the chapter "Theory" and the simulations run with the software HXSim that provides with a more real point of view of how the device will behave in actual situations.

Temp. air in [ºC]	40
Temp. air out [ºC]	25
Temp. CO_2 [ºC]	20

Table 4.2. Temperatures of the fluids in the evaporator

The initial values of the air temperature, shown in table 4.2, were decided by following [20] even though the outer temperature was finally reduced. For the CO_2, the value in [13] 16 ºC, was taken initially as optimal but, after the first calculations of the heat exchanger, it was seen that this temperature could be risen until the current value as the size of the heat exchanger was excessively and strangely small in comparison with actual devices.

Based on this the first calculations with the provided formulas were done with Excel and several values were changed in acceptable ranges of variation to find a good performance for the heat exchanger. The main goals for the design of the devices were:

- A good overall heat transfer coefficient
- The less number of tubes possible
- The smallest size of the heat exchanger possible

The first comparison was done only with Excel and the purpose was to check if an inline or a staggered configuration of the tubes in the heat exchanger was preferred regarding the criteria previously explained. This way, table 4.3 shows the comparison between some of the results obtained with one and the other combination, standing out the most important ones.

	Staggered configuration	**Linear configuration**
Total exchange area [m^2]	411,83	565,25
Area per tube [m^2]	0,350	0,333
Fin performance [%]	98,94	99,24
Air heat transfer coefficient [W/m^2·K]	84,76	58,43
Overall heat transfer coefficient [W/m^2·K]	70,02	51,01
Number of tubes	1177	1700
Depth of heat exchanger [m]	0,115	0,171

Table 4.3. Comparison values of the staggered and linear configuration

Once the kind of configuration was selected, the next step consisted on introducing this data in the HXSim and see how real the approach done was. In order to find an operation of the device that was valid and common between, the diagram shown in figure 4.1 was followed as method. The idea is to introduce the calculated values in the simulator and check if the actual heat exchanger is properly designed, oversized or undersized.

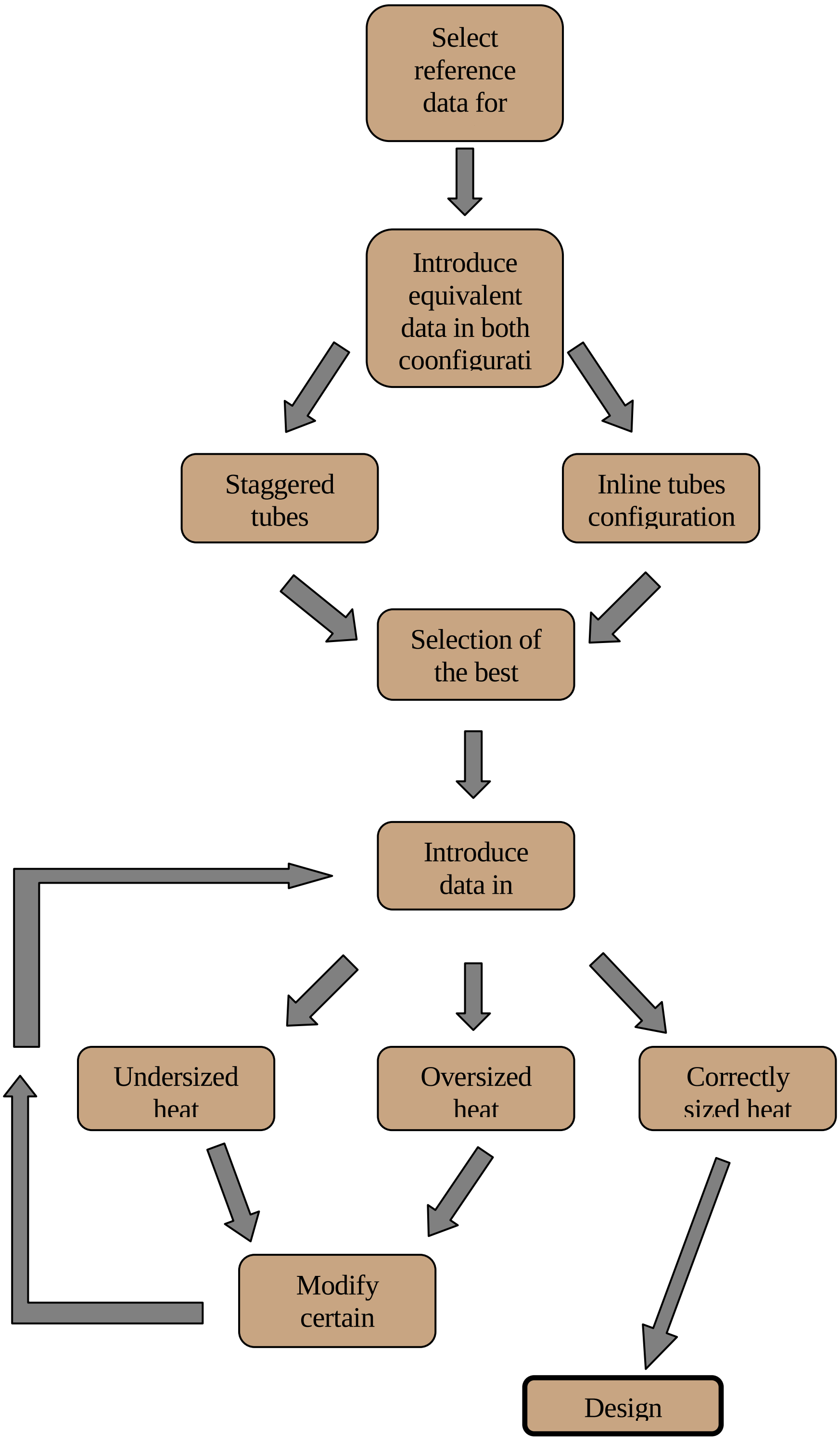

Figure 4.1. Diagram of the designing method followed (Source: Own design)

When introducing the first approach values calculated with Excel in the HXSim, it is find out that the system is oversized. The cooling capacity of the actual heat exchanger is way bigger than the one needed, which means a waste of money due to oversizing.

Needed cooling capacity [kW]	Simulated cooling capacity [kW]
312	396,73

Table 4.4. Comparison between desired and simulated value

This configuration has been set with the values shown in table 4.5.

	Initial configuration
Tube diameter [m]	0,0952
Horizontal pitch [m]	0,0961
Vertical pitch [m]	0,0211
Fin pitch [m]	0,0045
Number of tube layers	12
Tubes per layer	97
Depth of the heat exchanger [m]	0,115
Number of fins	1244
Airflow rate [m^3/s]	18,12

Table 4.5. Initial configuration of the heat exchanger

Thus, it is necessary to redesign the heat exchanger in order to find a cooling capacity that approaches the amount of heat rejected by the IT equipment. This way, several variations of key values of the heat exchanger were modified in HXSim like the number of tubes, air flow needed, distances between tubes... Finally, a non unique but accurate solution for the design of the heat exchanger was found. The configuration is shown in table 4.6 and also the main results of the simulation.

	Simulation results
Tube diameter [m]	0,0952
Horizontal pitch [m]	0,0961
Vertical pitch [m]	0,0211
Fin pitch [m]	0,0045
Number of tube layers [-]	9
Tubes per layer [-]	98
Length of the heat exchanger [m]	6,1
Height of the heat exchanger [m]	2,059
Depth of the heat exchanger [m]	0,08649
Number of fins [-]	1306
Airflow rate [m^3/s]	9
Coolant massflow rate [kg/s]	2,165
Cooling capacity [kW]	313,21
Output air temperature [ºC]	27,72
Air temperature difference [ºC]	12,28
Fan power [W]	35,68

Table 4.6. Results of the definitive simulation

Figure 4.2 shows how the dimensions of the profile of the heat exchanger are in scale, and how is its staggered disposition. Furthermore, the 3.23 one shows a general view of the heat exchanger in three dimensions.

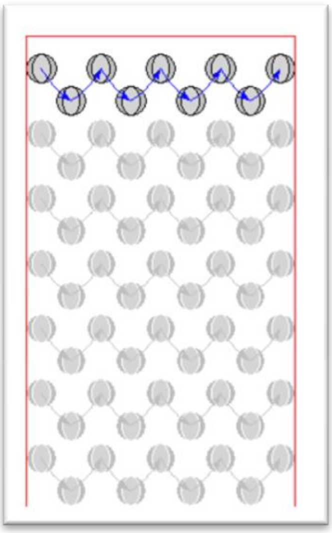

Figure 4.2. View of the tube distribution in the evaporator (Source: HXSim)

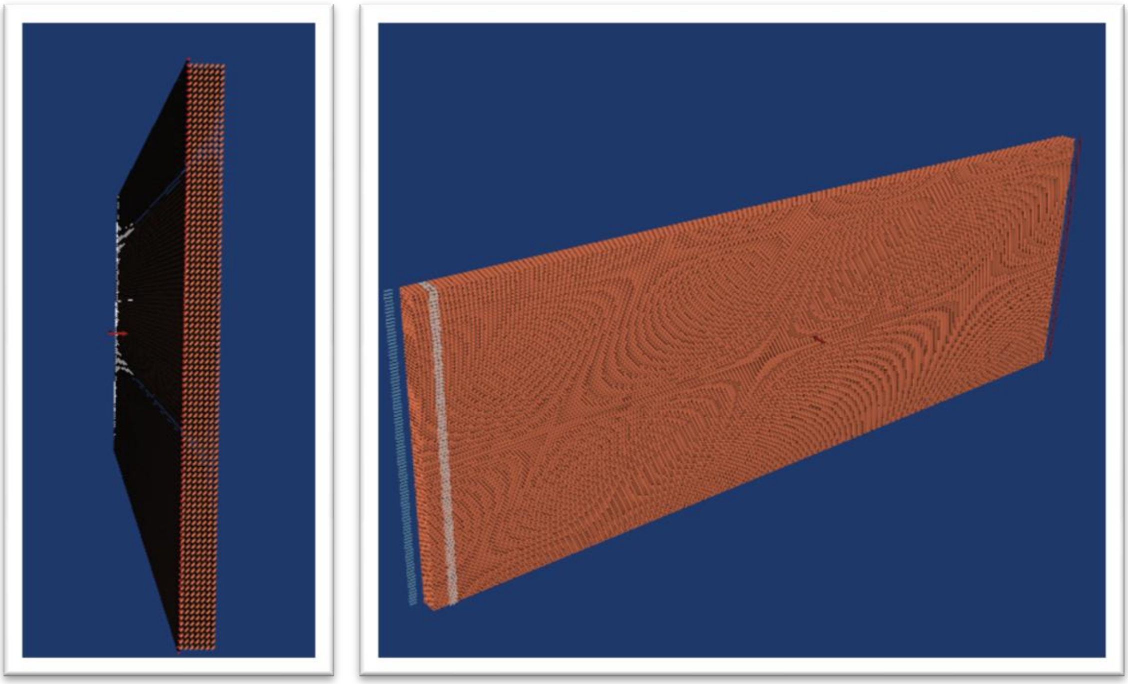

Figure 4.3. 3D view of the evaporator (Source: HXsim)

4.2.2. Fan airflow

By means of the formula 3.29 the mass flow needed to cool down one module of IT equipment was calculated. Moreover, with formula 3.13 which allows to know the free area between the tubes of the heat exchanger through which the air is going to pass, the velocity of the fluid was calculated. Table 4.7 show the initial values calculated with Excel for what will be the approach of the volumetric flow and the fan power needed.

ΔT [ºC]	15
Air mass flow for 1 evaporator [kg/s]	20,66
Air volume flow for 1 evaporator [m^3/s]	18,12
Maximum velocity [m/s]	4,76
Total air volume flow [m^3/s]	36,24
Total fans installed	12
Air volume flow per fan [m^3/s]	3,02

Table 4.7. Initial fan air flow calculated

However, after the simulation done, it was found than the air flow needed is way smaller than the previous calculated, which means variations in the CFM needed for moving the air and consequently in the size of the fan. Figure 4.4 shows how the design of the fans all together with the heat exchangers will be.

ΔT [ºC]	12,28
Air mass flow for 1 evaporator [kg/s]	9,775
Air volume flow for 1 evaporator [m^3/s]	9
Face velocity [m/s]	0,74
Total air volume flow [m^3/s]	18
Total fans installed	12
Air volume flow per fan [m^3/s]	1,5

Table 4.8. Definitive fan air flow calculated

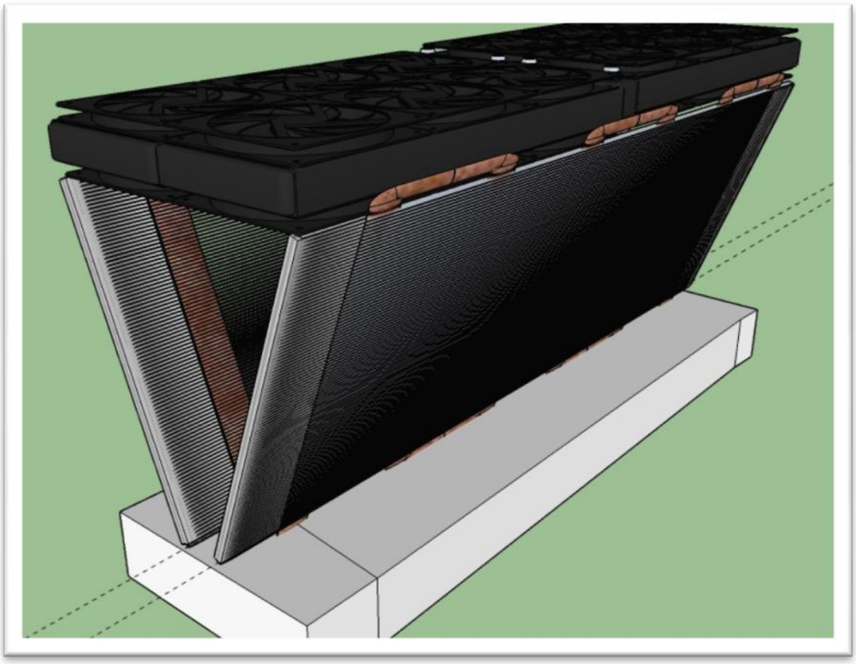

Figure 4.4. 3D model position of the heat exchangers together with the fans (Source: Own design)

4.3. Thermosyphon module

As it is said in the "Theory" section, the CO_2 in this module will be operating just with the change of state, so only its latent heat is going to be used for capturing the heat. Ideally, the cycle will be working as it is shown in figure 4.5, as there is no pressure input into the cycle.

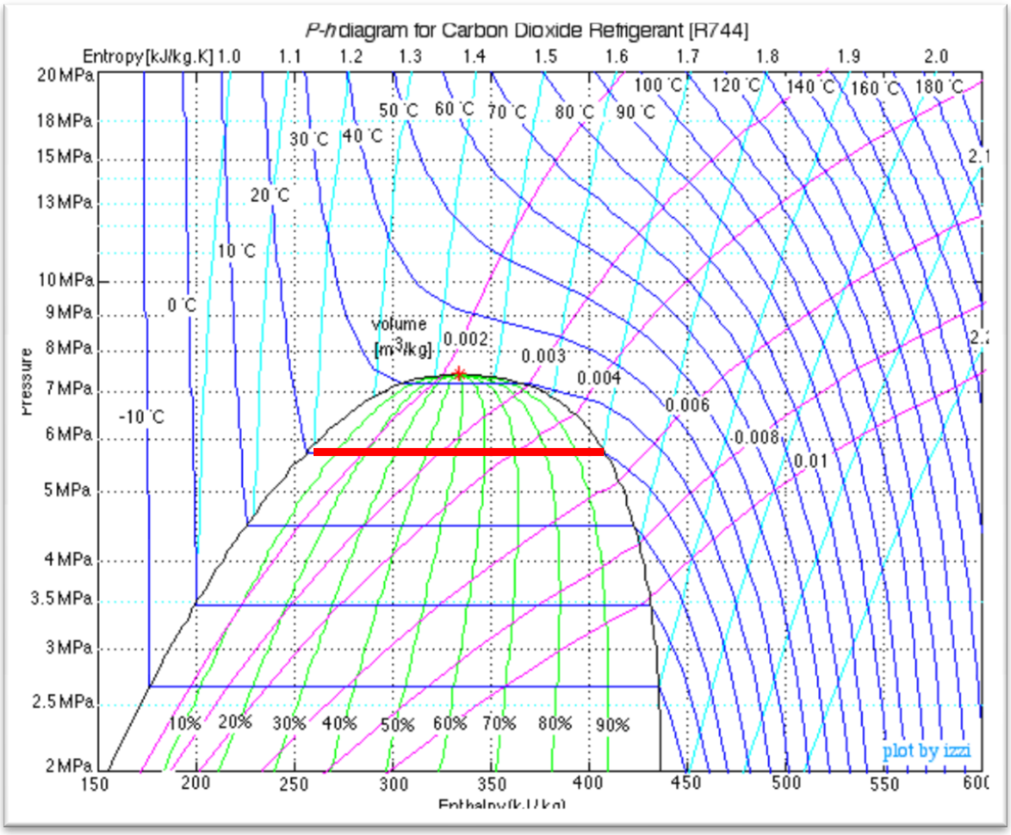

Figure 4.5. Ideal cycle of the thermosyphon loop (Source: Own design)

Obviously, this situation is ideal. The fluid losses pressure while circulating through the pipes, evaporators, condensers and other elements of the circuit. That is way is necessary in this case to design the circuit with a specific diameter for the tubes. As explained previously, formula 3.30 gives the pressure difference the system is able to develop thanks to the thermosyphon design, so by means of formula 3.32 it is necessary to find a diameter of the pipe that allows the circulation of the coolant without adding any extra pressure from a pump.

Through the Matlab code designed it is possible to calculate the pressure lost allowed and the diameter needed with accuracy. However, first of all it is necessary to introduce in the code some initial data. Part of it was extracted from [25] as shown in table 4.9.

CO_2 saturated liquid density at 20ºC [kg/m^3]	773,4
CO_2 saturated vapour density at 20ºC [kg/m^3]	194,2
Gravity acceleration [m/s^2]	9,81

Table 4.9 Initial data set for the tube diameter calculation

Then, it is also necessary to know the flow mass that the circuit has to handle. Knowing the amount of heat rejected by on IT module, it is possible to know how much flow is going to circulate through the pipes by using formula 3.31 and the information from table 4.10.

Total heat [kW]	624
CO_2 mass flow [kg/s]	4,103

Table 4.10. Mass flow needed for the calculations

The next step will be simulating an air cooler with HXSim that suits the module in order to handle and reject all the heat produced. The values in table 4.10 all together with the measures of the module where the coolers will be placed work as initial data for designing the heat exchanger. After the simulations, it was found out that an unique device was not able to cool down all the flow. Due to the fact that the simulator is not able to perform a cycle where only heat latent is used (as in this case with the CO_2), there were appearing problems due to pinch point in the heat exchanger. Thus, for solving this problem, 4 exchangers will be place in series in order to reject all the heat.

Figure 4.6 shows a scheme of how would be the connection and distribution of the systems. Furthermore, in table 4.11 the results feature of the device can be seen and below, the 3d model is shown.

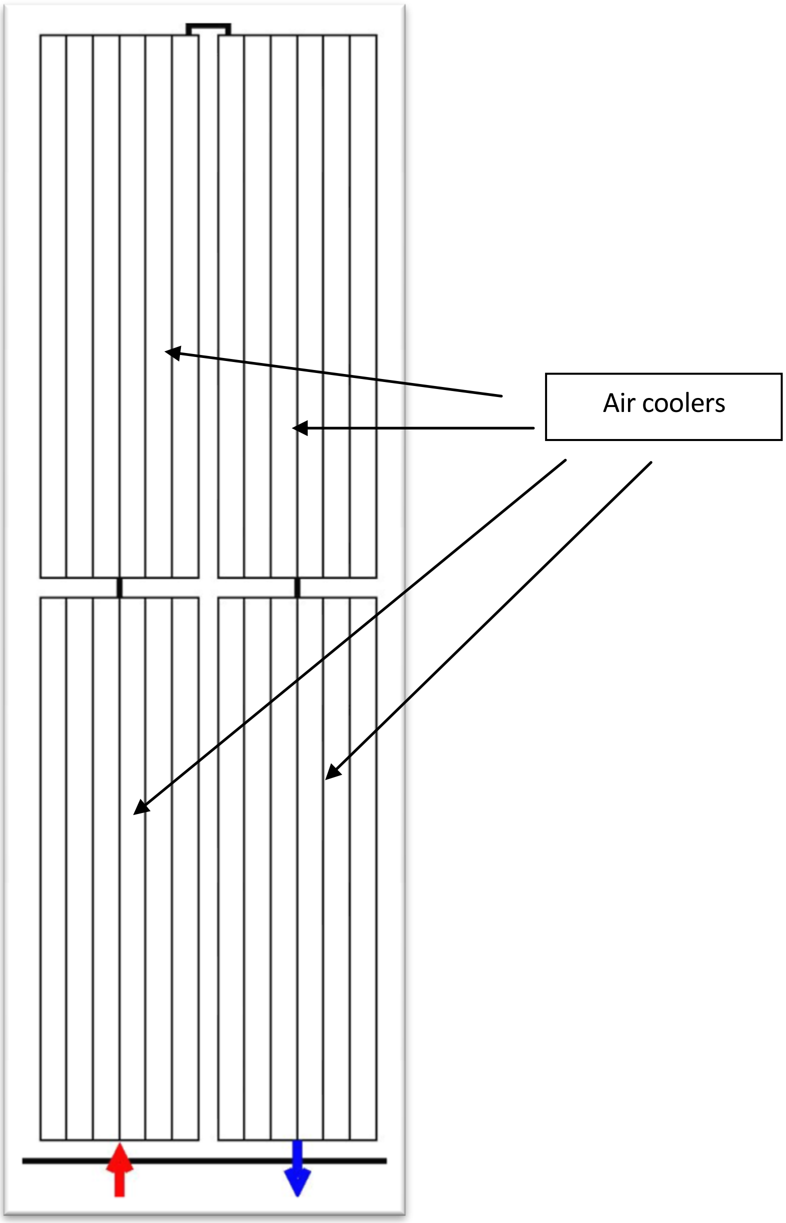

Figure 4.6. Connection of the air coolers in the thermosyphon module (Source: Own design)

Ammonia mass flow [kg/s]	4,1
Inlet refrigerant temperature [ºC]	20
Inlet refrigerant pressure [kPa]	5729
Air temperature [ºC]	0
Tube diameter [m]	0,00952
Horizontal tubes [-]	20
Vertical tubes [-]	20
Total tubes [-]	400
Core width [m]	3
Core height [m]	1,025
Core depth [m]	0,3
Horizontal pitch [m]	0,002
Vertical pitch [m]	0,05
Fin pitch [m]	0,0045
Fin thickness [m]	0,0004
Air volume flow [m^3/s]	220,82
Heat rejected [kW]	320,41
Air face velocity [m/s]	7,1
Air side pressure drop [Pa]	404,75
Fan power [kW]	8,1
Coolant pressure drop [Pa]	1009000

Table 4.11. Heat exchanger simulation results

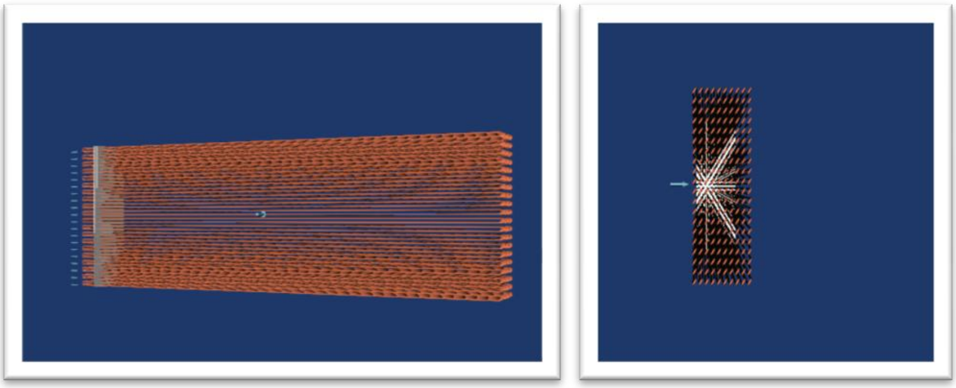

Figure 4.7. 3D view of the air cooler (Source: HXSim)

Finally, it is necessary to know the total length of the pipe what will allow to finally calculate the height losses. This is done by means of the formula 3.35 in which, in this case the length of the different devices in the circuit is going to be taken as an approximate value. Due to the fact that it is very complicated to know how the shape of the pipe would be in an actual situation, an approximation of the number of elements existing in the circuit is going to be done. In the case of the condenser and the evaporators, the simulations done with HXSim show a pressure drop value that is transformed in meters of height loss by means of formula 3.36.

	Number	Equivalent length [m]
Pipes	-	3,3
Small corners	4	12
Evaporators	2	122,5
Condenser	4	5200
Total	-	5337,3

Table 4.12. Height losses length data

This data allow, firstly, to calculate the difference of pressure existing in the system that will act as driving force by means of the formula 3.30, obtaining a value of **5681,95 Pa**.

Later, with formulas 3.32 to 3.34 the code calculates the pressure lost that is allowed in the system in terms of friction and the diameter of the tube needed. Obviously, the real diameter of the tube cannot be the one calculated exactly, as there is standardized measures for them. In this way, the real measure of the pipe, following [29], will be the immediately close bigger than the one computed.

Pressure difference [Pa]	5648,58
Computed diameter [m]	0,117
Standard diameter [m]	0,121

Table 4.13. Results after running the code

4.4. CO_2-Ammonia module

4.4.1. Space heating operation mode

Regarding the main loop of the module, the ammonia one, it has been designed with the software CoolPack. First of all, there are series of data that has to be introduced in the software, which are shown in table 4.14. Furthermore, the temperature differences in evaporator and condenser, that are also data that has to be included in the calculations, were calculated with formula 3.4 and are shown in the table below together with the temperature of all the fluids involved.

	Parameters
CO_2 thermosyphon loop temp. [ºC]	20
Inlet water temp. [ºC]	30
Outlet water temp. [ºC]	40
ΔT condenser [ºC]	14,42
ΔT evaporator [ºC]	10
Discharge temperature (T_2) [ºC]	60
Outlet temperature (T_8) [ºC]	20
Heat to reject [kW]	624
Isentropic performance [-]	0,7

Table 4.14. Summary of the temperatures involved

Once this data is introduced in the software, the calculation can be done. Picture 4.8 shows the Carnot cycle drawn in the diagram with the different thermodynamic states. Table 4.15, shows all the data for each of the these states.

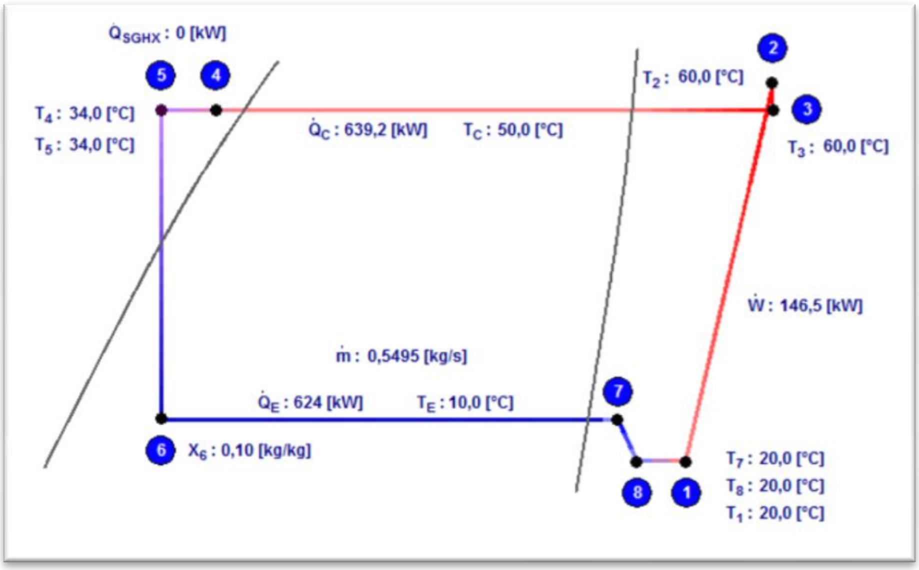

Figure 4.8. Ammonia Carnot cycle (Source: CoolPack)

Cycle point	Temperature [ºC]	Pressure [kPa]	Enthalpy [kJ/kg]	Density [kg/m^3]
1	20	606	1482,8	4,6
2	60	2062,7	1508,2	15,1
3	60	2036,3	1509,8	14,8
4	34	2036,3	346,5	588,9
5	34	2036,3	346,5	588,9
6	10	616,6	346,5	-
7	20	616,6	1482	4,7
8	20	606	1482,8	4,6

Table 4.15. Summary of the thermodynamic states

CoolPack also calculates the COP of the system and the pumping power needed, apart from other important parameters of the circuit. It has to be remarked that a compressor electric efficiency of 85 % was assumed.

Mass flow [kg/s]	0,553
Heating capacity [kW]	639,2
COP [-]	4,23

Compressor power [kW]	147,5
Electric compressor performance [%]	85
Electric power [kW]	173,53

Table 4.16. Summary of other important values of the cycle

4.4.2. Heat rejection operation mode (summer)

In this situation, the correspondent gas cooler was simulated with HXSim in order to check how the heat rejection system will be regarding the size of the heat exchanger, its capability for rejecting the heat and the electric consumption it will be needed for the fan power. Some of the data calculated in table 4.17 with the CoolPack is needed in order to introduce it in the HXSim for the simulations.

Ammonia mass flow [kg/s]	0,553
Inlet refrigerant temperature [ºC]	60
Inlet refrigerant pressure [kPa]	2062,7

Table 4.17. Summary of initial data for the simulation

Firstly it was found that, trying to cool all the mass flow in one condenser was not optimal. The fan power required was to big as the air pressure drop was highly increased when adding more tubes, so the software was not able to run the simulation. Thus, two different condensers were placed in the system so the cooling process would be done by separating the coolant flow and consequently reducing the amount of tubes of each condenser. Results of the simulations and the configuration of the heat exchanger are shown in table 4.18. This values are representative for one of the two machines, but both are equal.

Ammonia mass flow [kg/s]	0,275
Inlet refrigerant temperature [ºC]	60
Inlet refrigerant pressure [Pa]	2062,7
Air temperature [ºC]	30
Tube diameter [m]	0,00952
Horizontal tubes [-]	20
Vertical tubes [-]	20

Total tubes [-]	400
Core width [m]	2,2
Core height [m]	0,7175
Core depth [m]	0,44
Horizontal pitch [m]	0,002
Vertical pitch [m]	0,035
Fin pitch [m]	0,0045
Fin thickness [m]	0,0004
Air volume flow [m^3/s]	10
Heat rejected [kW]	320,41
Air face velocity [m/s]	7,9
Air side pressure drop [Pa]	708,04
Fan power [kW]	7,1
Coolant pressure drop [Pa]	16180

Table 4.18. Results after the simulation

A picture of the actual size of the heat exchanger is shown in figure 4.9 and below it, so is the idea for the distribution of the cooling devices inside the module.

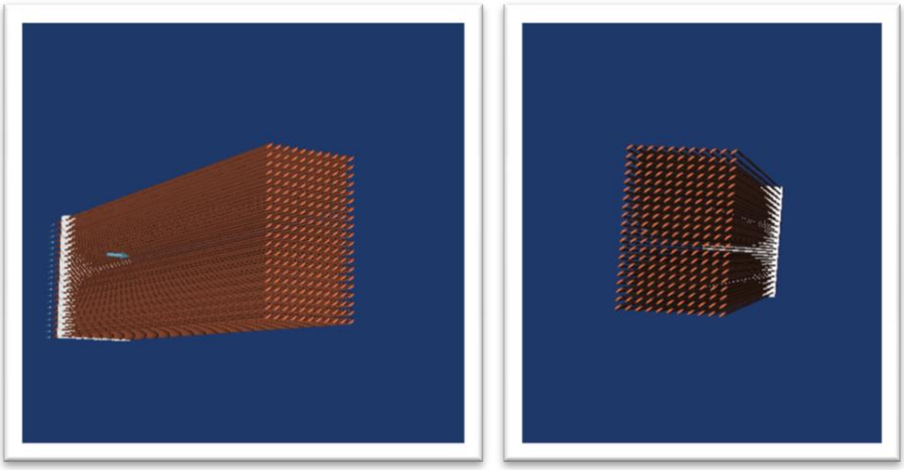

Figure 4.9.3D view o the ammonia air coolers (Source: HXsim)

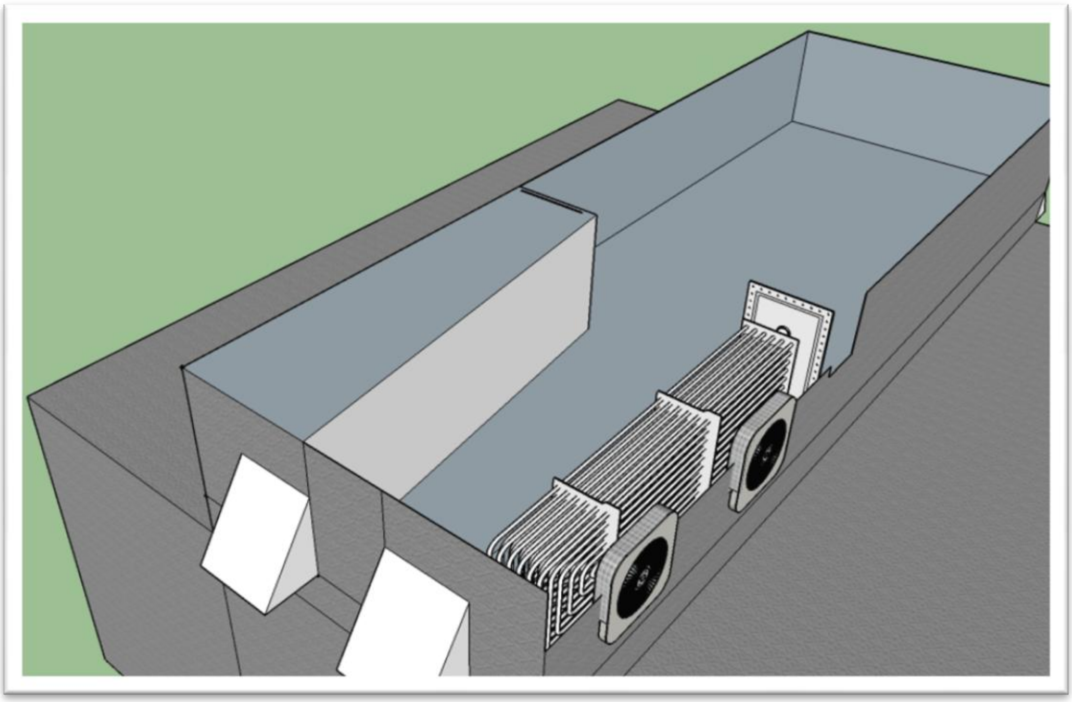

Figure 4.10.3D view of the distribution fo the air coolers inside the module (Source: Own design)

4.4.3. Heat rejection operation mode (winter)

As explained in the prior chapters, the CO_2 loop is the one connected to the main evaporators capturing the waste heat and afterwards delivering it to the ammonia one. As in the configuration 1, this loop is only going to work using the latent heat of the CO_2, so there is no need to add pressure to the system. Thus, the work principle in this situation is again the thermosyphon loop, which doesn't need pumping power to work. The only difference with the previous case is that a few meters more of pipes are going to be need for arriving until the thermosyphon module, which will be place a little higher.

	Number	Equivalent length [m]
Pipes	-	3,8
Small corners	4	12
Evaporators	2	122,5
Condenser	4	5200
Total	-	5337,3

Table 4.19. Summary of equivalente lengths

After running the Matlab code, the results obtained were almost the same as in the thermosyphon case due to the insignificant increment in the length, so values in table 4.13 are also valid in this case.

4.5. CO_2-CO_2 module

4.5.1. Water heating operation mode

The second loop, however, is different in this occasion. CO_2 is, as commented previously, working in transcritical cycle so it has specific operation conditions and properties. However, as in the other case it is necessary to set some initial data for running the program. This data is shown in table 4.20.

Outlet condenser temperature (T_4) [ºC]	20
Evaporator temperature [ºC]	10
ΔT evaporator [ºC]	10
Discharge temperature (T_2) [ºC]	100
Outlet temperature (T_8) [ºC]	20
Condenser pressure [MPa]	8,5
Heat to reject [kW]	624
Isentropic performance [-]	0,7

Table 4.20. Summary of initial data for calculations.

Again, as in the previous case, the temperatures of the other working fluids have to be set. The table below shows them.

	Temperature [ºC]
CO_2 first loop	20
Inlet water	10
Outlet water	90

Table 4.21. Working fluids temperatures

Here is where the interesting facts of the CO_2 transcritical cycle can be seen. If the goal of the system was to reheat water, where the inlet water temperature would be higher (let's say 40ºC) , its performance would decrease drastically as values of table 4.23 show.

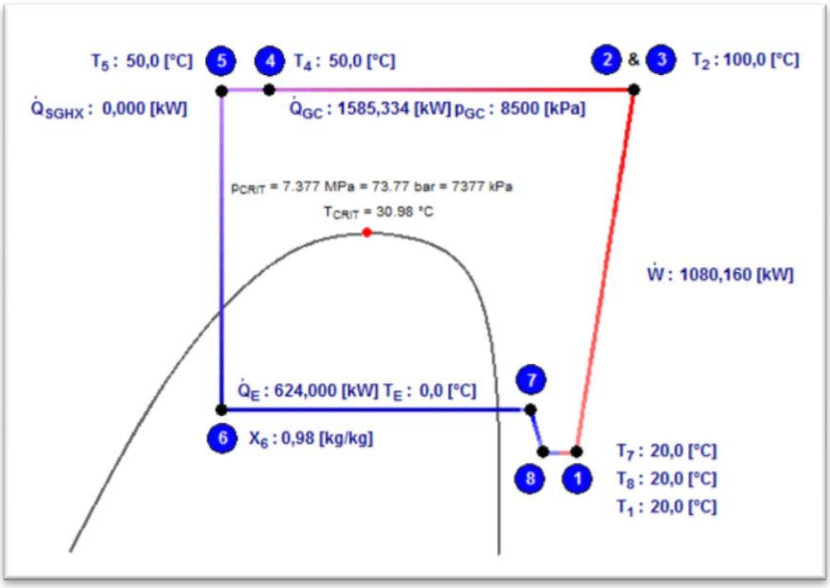

Figure 4.11. CO_2 bad performing transcritical cycle module (Source: CoolPack)

Cycle point	Temperature [ºC]	Pressure [kPa]	Enthalpy [kJ/kg]	Density [kg/m^3]
1	20	3467	-45,1	79,8
2	100	8500	9,3	152,5
3	100	8500	9,3	152,5
4	50	8500	-80,9	248,9
5	50	8500	-80,9	248,9
6	0	3485	-80,9	-
7	20	3485	-45,4	80,4
8	20	3467	-45,1	79,8

Table 4.22. CO_2 bad performing cycle thermodynamic states

Table 4.23 shows the rest of the calculations done and important parameters calculated with the software.

COP [-]	0,58
Coolant mass flow [kg/s]	17,58
Pumping power [kW]	1080,16

Pump electric performance [%]	85
Pump electric power [kW]	1270,78
Heating capacity	1585,33

Table 4.23. CO_2 bad performing cycle results

This values show that the circuit is not designed for the proper application. The COP value is lower than 1, which means that the circuit consumes more electricity that cooling power has. Furthermore, the refrigerant flow is very high which also requires a very high pumping power and makes the cycle not efficient at all. Thus, it is then basic that the system operates in a mode that heats up totally the water, from the tap until its maximum value. This big temperature gradient will make the system perform properly.

The correct Lorentzen cycle is shown in figure 4.12, as well as the thermodynamic states presented in table 4.24.

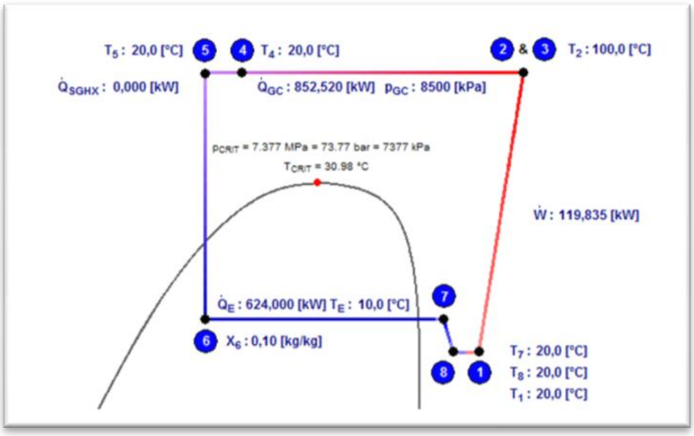

Figure 4.12. CO_2 well performing transcritical cycle module (Source: CoolPack)

Cycle point	Temperature [ºC]	Pressure [kPa]	Enthalpy [kJ/kg]	Density [kg/m^3]
1	20	3467	-62,7	116
2	100	8500	9,3	152,5
3	100	8500	9,3	152,5
4	20	8500	-261,1	835,8
5	20	8500	-261,1	835,8
6	10	3485	-261,1	-
7	20	3485	-63,2	117
8	20	3467	-62,7	116

Table 4.24. CO_2 well performing cycle thermodynamic states

Table 4.25 shows the rest of the calculations done and important parameters calculated with the software.

COP [-]	5,207
Coolant mass flow [kg/s]	3,15
Pumping power [kW]	119,84
Pump electric performance [%]	85
Pump electric power	141
Heating capacity [kW]	852,52

Table 4.25. CO_2 well performing cycle results

In this case it can be seen that the performance of the system is way better. The COP has increase to a valid value, the CO_2 mass flow has been reduces to a acceptable value as well as the pumping power.

4.5.2. Heat rejection operation mode (summer)

As in the correspondent case of the ammonia module, the gas cooler was also simulated with HXSim in order to check how the heat rejection system will be in this module. Knowing the previous problems with pressure drop in the heat exchanger due to very high number of tubes, the same methodology was followed and two heat exchangers were placed. All the values related with their design are shown in table 4.26.

CO_2 mass flow [kg/s]	1,615
Inlet refrigerant temperature [ºC]	100
Inlet refrigerant pressure [Pa]	8500
Air temperature [ºC]	30
Tube diameter [m]	0,00952
Horizontal tubes [-]	20
Vertical tubes [-]	20
Total tubes [-]	400
Core width [m]	3
Core height [m]	0,718
Core depth [m]	0,40
Horizontal pitch [m]	0,002

Vertical pitch [m]	0,035
Fin pitch [m]	0,0045
Fin thickness [m]	0,0004
Air volume flow [m^3/s]	15
Heat rejected [kW]	426,42
Air face velocity [m/s]	7,56
Air side pressure drop [Pa]	726,84
Fan power [kW]	10,9
Coolant pressure drop [Pa]	184550

Table 4.26. CO_2 air gas cooler parameters

A picture of the actual size of the heat exchanger is shown in figure 4.13 and below it, so is the idea for the distribution of the cooling devices inside the module.

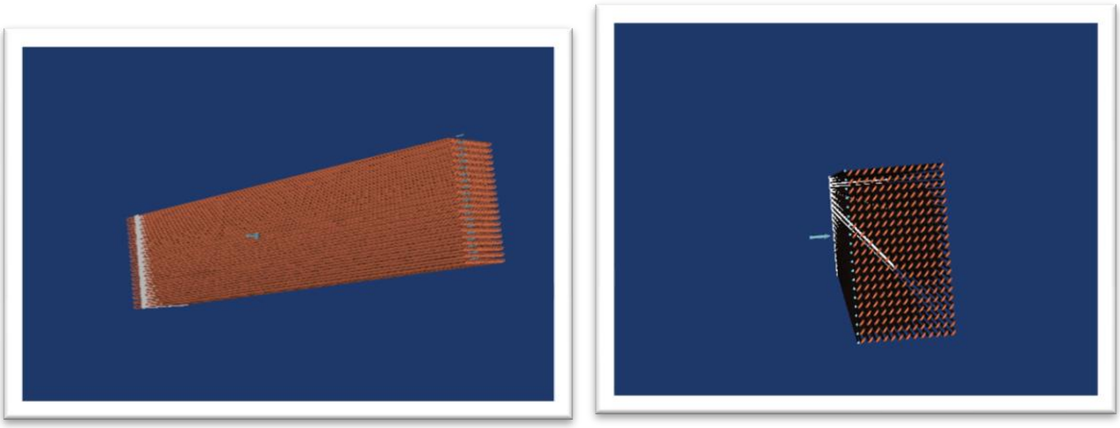

Figure 4.13. 3D view of the CO_2 air coolers (Source: HXSim)

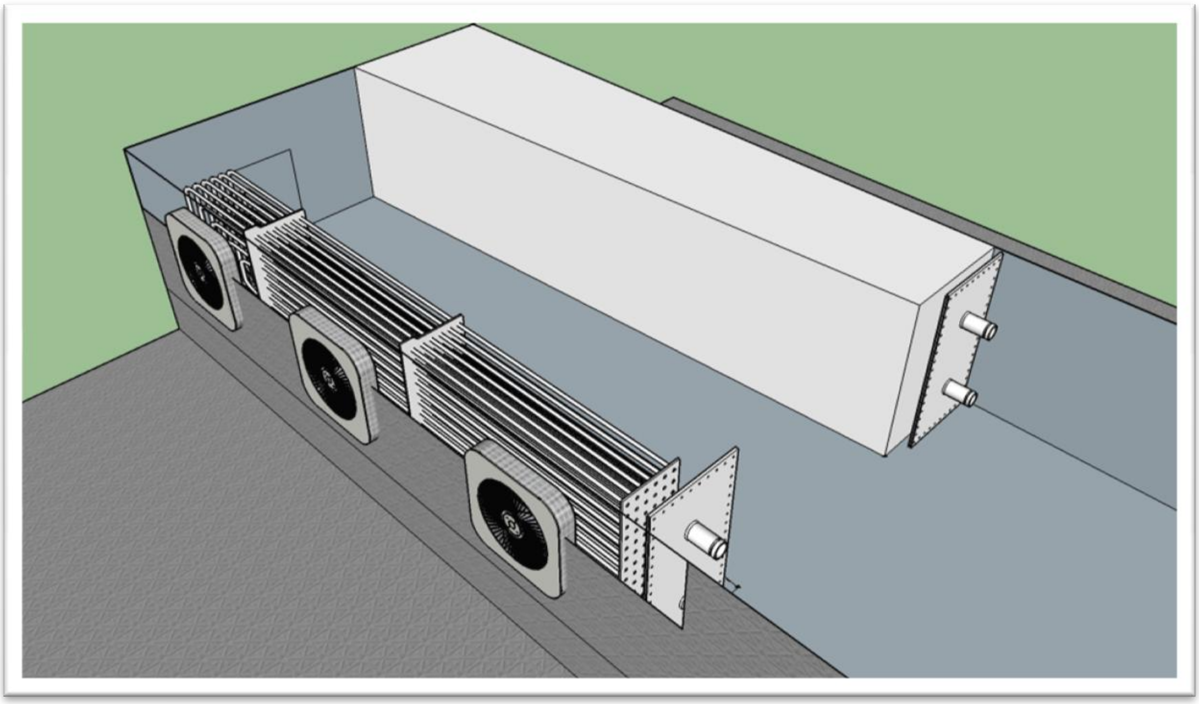

Figure 4.14. 3D view of the distribution of the designed coolers(Source: Own design)

4.5.3. Heat rejection operation mode (winter)

The configuration of this module is very similar to the previous one. The pumping power of the first module is the same as in the prior case, as all the cycle settings are the same. Then, values in table 4.13 are also valid in this case.

4.6. Energy analysis

Regarding the performance of the system, is interesting to develop an small energy analysis which indicates the main features of the system in order to have a general view of the performance of the data center. First of all, then, it can be seen in table 4.27 a summary with the electric consumption of the data center and the heat rejection regarding the configuration of the cooling system.

	Configuration 1	Confguration 2	Configuration 3
IT power [kW]	624		
Heat rejected [kW]	624	639,2	852,52

Table 4.27. Consumption and rejection for each configuration

In this way, an analysis of the electric consumption of the data center, also depending on the configuration that is operating is shown in table 4.28. Basically, this consumption analysis takes into account the energy requirements of the main elements in the data center as, obviously the IT equipment, fans and pumps. Other less important aspects are lightning are not considered.

Description	Configuration 1	Configuration 2	Configuracion 3
Fans cooling module [kW]	0,07	0,07	0,07
Fans thermosyphon module [kW]	32,4	32,4	32,4
Fans air coolers [kW]	-	14,6	21,8
Pump power [kW]	-	172,35	141

Table 4.28. Electric energy analysis of each configuration

This analysis is carried out calculating the PUE of each configuration and it can be seen in the table below. The main purpose is to have values that can be compared with the PUE references existing.

	Configuration 1	Configuration 2	Configuration 3
PUE rejection summer	-	1,299	1,261
Pue rejection winter	1,052	1,052	1,052
PUE heat reuse	-	1,276	1,226

Table 4.29. Summary of the PUE's for each configuration

Furthermore, it has also been done an approach of the amount of residential households that can be supplied with both configurations, space heating and water heating production, taking into account Nordic energy requirements shown in table 3.7.

4.6.1. Space heating

First of all, the study for the amount of dwellings that can be supplied with the amount of heat produce for space heating is going to be done. Knowing the amount of energy rejected by the system and the consumption per square meter, the m^2 that can be warmed up can be calculated and also the number of dwellings supplied. Table 4.29 summarize the values calculated.

Heat from data center [kW]	639,2
Area warmed up [m^2]	24558,74
Dwellings supplied [-]	213

Table 4.30. Study of the amount of households supplied

4.6.2. Water heating

The same process is followed for calculating the amount of water for obtaining the values shown in table 4.30.

Heat from data center [kW]	852,52
Area warmed up [m^2]	84015,85
Dwellings supplied [-]	730

Table 4.31. Study of the amount of households supplied

5. ENVIRONMENTAL IMPACT STUDY

The main purpose of this project is, ultimately, trying to reduce the environmental impact of data center in every manner possible, regarding its cooling issues. The fact that the consumption of this kind of IT equipment is growing and they are being more demanded each year cannot be changed, as data centers have shown to be an essential part of telecommunications world nowadays. However, there are a lot of things that can be done in order to make them more sustainable and energy efficient.

First of all, just the idea modular designed leads to energy savings in the system. Pre-engineered modules make easier the fabrication process of the system which is also translated in a process that consumes less energy, and they are also very easy and quick to install which also reduces energy consumption in this part of the process. Furthermore, its mobility makes that they can be placed anywhere. In this way, usually modular data centers are placed not far from electric stations so the electric connection is close and the losses in the cable are minimum.

Modular design also has other advantages that influence lately the energy performance. The fact of having every device integrated and tight in the modules, leads to efficiency gains both in power and cooling [19], and there is the possibility of using energy management platforms which will improve the general performance of the system as a whole. Modular data centers have usually more density than the traditional ones, being able to have 20000 W and more per cabinet [19], as in the case explained. Thus, this allows more precise airflow for cooling the equipment which is finally translated in efficiency gains in the whole system.

Once the benefits regarding environmental impact of the modular design of data centers are explained, is moment to focus in the technologies applied in this system regarding the cooling.

Historically, most used fluids in cooling applications were, for example, R-134A or R-410A, synthetic working fluids with acceptable thermal properties. The problem with this kind of coolants comes when the concept of Global Warming Potential (GWP). GWP is a measure of the relative amount of heat that a determined greenhouse effect gas can trap in comparison with a reference gas, usually CO_2, so it can be deduces then that 1 is the GWP value for CO_2 as shown in table 2.2. This means then than a coolant that is has a specific value for the GWP, is releasing to the atmosphere that value times CO_2. Obviously, the concept of GWP is relative. If a determined coolant has a slightly bigger value in the parameter but is easier to remove from the atmosphere, then its damage is not that big. However, the traditional coolants mentioned before have way too big GWP values, as it can be seen in table 2.2, so it was necessary to find new coolants that suits the actual requirements regarding global warming issues. In this context, both CO_2 and ammonia appear as global worming respecting coolants. In the case of CO_2, as said before, its value is 1, and even better is the ammonia with a value of 0, having no global warming effect. Under this circumstances, it can be said that the cooling system of this data center is totally environmentally friendly when it comes to avoid greenhouse effect.

Keeping on talking about the energy saving technologies applied to the data center, it is time to stand out the high performance heat pumps. Heat pump technology allow, with a very low electric consumption, to absorb the heat for a determined heat load by means of the use of different types of working fluids.

	COP
CO_2 heat pump	5,207
Ammonia heat pump	4,23

Table 5.1. COP value for each heat pump system

Table 5.1, shows briefly the results obtained regarding the heat pumps performance. The CO_2 one is able to reject 5,207 thermal kW per each electric one, while the ammonia one rejects 4,23. This COP values express the idea that the cooling systems are well designed as, furthermore, when checking the pumping power and calculating the PUE of the data center, it can be seen in table 4.28 that its values are very acceptable regarding the range established in [30]. Moreover, the strategic decision of using a specific coolant for each different application also leads to energy savings. In previous sections it has been seen the importance of the temperature level with respect to the coolant used, as the selection of a different fluid would have lead to important efficiency losses.

Finally, what could be the most important fact regarding environmental impact and efficiency of the system is the fact of having modules designed for making use of the waste heat. This feature leads to very big energy savings as, it can be seen in tables 4.30 and 4.31 from the energy analysis, the data center is able to feed several residential buildings in a year in both space heating and water heating. This amount of heat would be produce in other situations either by electric boilers that consume electricity, which in most countries comes mostly from non renewable energy sources, or by other kind of thermal systems, like biomass boilers which, even though they have a smaller CO_2 footprint, also produce greenhouse effect gasses.

6. DISCUSSION

There are several different points regarding the design of the data center that are interesting to stand out. The main idea that can be extracted from it is the concept of trying to design in a modular way, which leads to the idea of reducing everything to the minimum space for getting all the advantages of a modular design, but also keeping a good performance in the system. This concept can be expressed saying that modular data centers are a big and very complex optimization problem where there are a big amount of variables that depend on each other.

6.1. Heat exchanger in cooling module

With respect to the main heat exchangers placed in the cooling module, there are several aspects that can be commented. First of all, it has been seen how important is the distribution of the tubes in the heat exchanger for its design, performance and economic cost. Under same conditions, a heat exchanger designed with a staggered configuration presents a way better performance with less tubes and occupying less space, as shown in table 4.3, for what there are several things to talk about. This distribution of the tubes makes that each layer of the heat exchanger is closer which can be lately translated in less depth in the heat exchanger. For other applications, this might not be an important feature to stand out, but when designing in modular way, as commented, space is a matter of importance. Furthermore, this configuration also increases the heat transfer. As it can be seen with formulas 3.9 and 3.10, the Nusselt number for the two types of configurations is different, being better in the case of the staggered one. This, affects directly in the calculations of the heat transfer coefficient of air, which then has also direct influence in the overall heat transfer coefficient, for finally calculating the exchange area.

After remarking the results obtained by comparing staggered and linear configuration, it is time to compare the values calculated with the formulas extracted from [25] with the solution find in the simulations. The comparison between the values show that the formulas are not really accurate and they do not represent properly the performance of the heat exchanger, even though they act as a good starting point. The most remarkable point that shows it is that, with a similar size heat exchanger as tables 4.5 and 4.6 shown, with 3 times less air flow through the heat exchanger the same amount of heat is captured form the IT equipment. Actually, when modifying the variables of the heat exchanger in the Excel calculations it was seen that the variations were very linear and did not really represent how the heat transfer in a heat exchanger is. HXSim is able to, by using an iterative method, find a representative solution for the problem, as in takes into account more complex features happening in the heat exchanger, that is complicated to compute with simple calculations in an Excel file.

6.2. Thermosyphon

The results obtained regarding the system operating with the thermosyphon loop only are satisfactory. The circuit is able to work by itself without any input from pumping energy. This, as it can be seen in table 4.29, has a very positive effect on the electric energy consumption when the configuration 1 (only rejecting the heat to the environment) is set. As electricity is only needed for running the fan, it can be seen in the same table that the PUE is very low. This fact is important in this situation due to a important reason: as in this situation the waste heat is not being reuse so there is not energy saving in that way, it is necessary to have a system that has minimum impact regarding energy consumption.

It is important also to stand out the performances of the air coolers. Due to limitations of the software is not possible to simulate the system as desired, as it does not allow to develop a system where the CO_2 only uses the latent heat, so when trying to cool down the correspondent amount of heat, pinch point problems were appearing in the tubes. Thus, the solution that was developed consists basically in designing one fourth of the total amount of heat for, later install four of them in series so the cooling process is done in 4 stages. It might not be optimal, but the solution approaches to what was optimally desired for this cooling system.

6.3. CO_2-Ammonia module

Since the initial idea, the flexibility of this module had been increased substantially. At the beginning only a system that is able to drive the heat until a water chiller for its future use was though, but finally it was found that this module should also be able to reject the heat just in case the space heating was no needed, and to reject it in every situation. For this reason, the solution found was optimal. The connection with the thermosyphon loop allows this rejection when the outer temperature is lower than the 20ºC that the CO_2 is working at. However, the introduction of a gas cooler that is connected in the ammonia loop allow also to a high outer temperature heat rejection. Basically, this idea increases the range of use of the module as it can be useful under several climate conditions, which means different countries and seasons of the year.

Regarding the ammonia loop performance, the optimal properties that ammonia has for this kind of application were checked. With low mass flows, as it can be seen in table 4.16, provides a big capacity for absorbing the heat. Furthermore, ammonia presents a good COP value for the temperatures under which the system is working, as shown in the same table. However, an interesting fact regarding the performance of the system when changing the temperatures of the system was seen when establishing the thermodynamic states of the cycle.

- When increasing the condenser temperature, the COP of the system suffers a decreasing on its value, even though the temperature gradient between ammonia an water would be bigger. Given an amount of heat that has to be rejected, when increasing the temperature of the condenser, the pumping power increases to which observing formula 3.37 shows that will decrease the COP, and the energy consumption will be bigger per unit of heat extracted.
- Following the same principle, when decreasing the temperature in the evaporator, the COP of the system suffers the same fact previously commented, as the pumping power is increased but not the same way the amount of heat that has to be rejected.

As it can be seen then, ammonia heat pumps are better for applications with a low and medium temperature level. Space heating with radiant floor requires a maximum level of approximately 40 ºC, which suits the technology capacities. If the water heating module or a district heating one (that requires higher temperature level) had been developed with an ammonia heat pump and also taking into account the evaporator temperature allowed, the performance of the system would be way worst.

6.4. CO_2-CO_2 system

The analysis or discussion of the first loop of this module is already explained in the previous section, as the operation mode is the same as with the ammonia heat pump. However, it is important to explain the performance of the Lorentzen or transcritical cycle, presented in the "literature review" section.

As it can be seen in table 4.25, the results obtained with CoolPack are very positive. The system presents a very high theoretical COP which means very low pumping power for the amount of heat extracted from the data center, fact that can be translated in a lower PUE of the data center.

Comparing tables 4.23 and 4.25 it can be seen how the transcritical cycle behaves regarding its temperatures. As the water which is going to be warmed up has temperature levels of 40 in the inlet and 90 in the outlet, a temperature gradient in the condenser of 10 degrees was taken into account, so 50 degrees in the outlet should be enough as established in table 4.22. However, in can be seen that with this configuration the system has a very bad performance. Pumping power is so high that it results in a very low COP, even under 1. That means that the system is actually consuming more energy that it is generating. This problem can be solved by decreasing the outlet temperature. This fact reduces drastically the amount of mass flow needed in the cycle which is also translated in a very low pumping power in comparison, what leads to a high COP and high performance.

Furthermore, it was also possible to see that a higher temperature in the evaporator helps to increase the performance in the system too. Taken into account that the temperature of the CO_2 is 20ºC and a temperature gradient of 10 is desired in the evaporator, the temperature limit that the second CO_2 loop has in the evaporator is 10 ºC, as shown in table 4.20. Then, it can also be seen in this table how the performance of the system changes when rising this temperature from 0 to 10 ºC, even though the temperature gradient in the evaporator is bigger in the second case.

Regarding the air coolers in this configuration, the only problem found was a slightly difference between the coolant mass flow needed. When simulating the exchangers, an extra amount of mass flow was needed in comparison with the calculations made with CoolPack. However, the differences is not big and the results can be taken as acceptable.

6.5. Energy analysis

It is very interesting to see the energy savings that can be produced by implement all the different kinds of cooling systems that can be applied to this system. If for example a normal heat pump with an air cooler had been implemented instead of the thermosyphon loop, the global performance of the configuration would have been very bad. The important idea of this configuration is that even though all the heat rejected is not reused, the system does not almost consume extra electricity apart from the one of the IT equipment (the PUE value is 1).

With respect to the energy reuse, it is important to have a very general vision of it. Data centers are a necessity nowadays and they are going to exist either they have other uses or not, but it is important to see the good consequences of implement system that are able to have a positive environmental impact. The fact that they are increasing a lot in number year by year and its development in a modular way which allows flexibility (they can be placed almost everywhere), all together with the introduction of cooling systems that allow reusing of the waste heat shows to be a very energy solution with a very good environmental impact. Such a small infrastructure, that almost does not take space, is able to provide space heating for 213 or water heating for 730 dwellings in a year. If future data centers are placed close to new built neighbourhoods or connected to district heating systems, the amount of energy saved could be increased exponentially.

7. FUTURE IMPROVEMENTS

Due to the fast develop introducing data centers in nowadays world and the growing existing trend in power density and number of utilities, it is important that its whole design including the cooling system develops as fast and uses the last existing technologies.

First of all, the whole size and design of the physical structure of the data center could be optimize. In this report, many of the measures of the container were design taking into account good assumptions for keeping the needed spaces, however, the solution is probably no totally optimize. An optimum solution in this aspect would lead to savings in money regarding the materials use for building the data center. Together with the optimization of the size of the container, a Computational Fluid Dynamics (CFD) analysis of the thermal behaviour could be done in parallel to its design. CFD allow to know precisely how is the behaviour of the heat transfer in the whole system, together with the temperatures reached in the system and the heat flow both in the air and in the coolants. When everything is so tightened built as in a data center is, little modifications in the measures of the container or the size of the elements inside the system can be translated in important variations in the heat flow behaviour, which potentially can lead to better performances of the system, or savings due to reduction of, for example, number of tubes in a heat exchanger.

Regarding the cooling options, as it can be seen in the literature review, there are infinite opportunities for improving the system in future modular designs. In this case, a air cooling solution was applied due to cost issues in the system and simplicity of the model, but as explained before, due to the better thermal properties of liquids in comparison to air, this type of solution would get more important in future design of data centers, as for the moment they are also more expensive to implement.

Finally, another flashy idea that can be introduced in data center for improving the energy performance and mainly the environmental impact is the use of renewable energies for electrically feeding the equipment. If the modules are placed close to a electric substation fed by renewable sources like wind mills or hydroelectric power, or solar panels are directly connected to be the electric supply, it could be developed a "Zero Energy Data Center" as all the consumption of the system would come from green sources.

Conclusions

Lets conclude by saying that data centers are the future not only when it comes to cloud computing, data store and processing, etc. but also they shown to be important source of renewable energy. This small technological isles have standed out thanks to their multifunctionality as they are able to provide way more than just computing services. The amount of energy that can be reused from them can be an important source of savings regarding electric consumption in the world if used properly in the next years. Results in this document show, roughly but significantly, how can be the impact this systems have if the latest cooling and engineering technologies are applied on them and how real is the possibility of drawing all its potential as renewable energy source.

Economic analysis

This economic study is focused in the repercussion that such a system has for the dwellings supplied, thus, money savings that go together with the heat pump technology are going to be calculated in a scenario where that technology is compared with a system that produces the same amount of heat with an electric boiler.

7.1.1. Savings in space heating

First of all, some initial data has to be set for starting the calculations.

Wlectric boiler efficiency [-]	0,9
Ammonia heat pump cop [-]	4,23
Electricity price [NOK/kWh]	0,8
Number of dwellings [-]	230

Table 0.1. Initial data for economic calculations

For the electric boiler efficiency, a common assumption was made, as nowadays this kind of systems have high performances. Regarding the energy price, [31] was checked in order to know an approximate value with which calculations can be made. Thus, following is shown a comparison table with the different energy and money values calculated with the formulas below.

$$C_{electric\ boiler} = \frac{E_{produced}}{\eta_{\text{boiler}}} \quad \textbf{(Eq. 0.1)}$$

$$C_{electric\ heat\ pump} = \frac{E_{produced}}{COP \cdot \eta_{compressor}} \quad \textbf{(Eq. 0.2)}$$

	Electric boiler	Heat pump
Energy produced [kWh]	5599392	
Electric consumption [kWh]	6221546	1557333
Cost [NOK]	4977237	1245866
Cost [€]	524053,3	131177,3
Cost per house [NOK]	23367	5849
Cost per house [€]	2460,34	615,9
Saving [NOK]	17518	
Saving [€]	1884,5	

Table 0.2. Results of the calculations

7.1.2. Savings in water heating

The same method is follow to perform the calculations regarding money savings in water heating. Thus, table 0.3 shows the initial data taken into account and 0.4 the results of the computes.

Wlectric boiler efficiency [-]	0,9
Ammonia heat pump cop [-]	5,207
Electricity price [NOK/kWh]	0,8
Number of dwellings [-]	713

Table 0.3. Initial data for economic calculations

	Electric boiler	Heat pump
Energy produced [kWh]	5599392	
Electric consumption [kWh]	6221546	1557333
Cost [NOK]	4977237	1245866
Cost [€]	524053,3	131177,3
Cost per house [NOK]	23367	5849
Cost per house [€]	2460,34	615,9
Saving [NOK]	8149	
Saving [€]	858,1	

Table 0.4. Results of the calculations

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Annex A

A1. Matlab code for height losses

```
%%Pedro Basanta Franco: Height losses


clear all
close all
clc

%Measures

l=5337;
dh=1;

%Fluid properties
ro_liq=773.4;
ro_vap=194.2;

ro=ro_liq;
mu=68.07e-6;

%Limit losses

dp_limit=(ro_liq-ro_vap)*9.81*dh;

%Flow

m_dot=4.103;
Q=m_dot/ro_liq;

dp_section=1;


 p=1;

 dpl=zeros(p);
 Diam=zeros(p);
for D=0.01:0.0001:0.14


%Velocity

c=Q/(pi*(D/2)^2);


%Reynolds
Re=(D*c*ro)/mu;

%Rugosity

eps=0.0024e-3;

%inital f
```

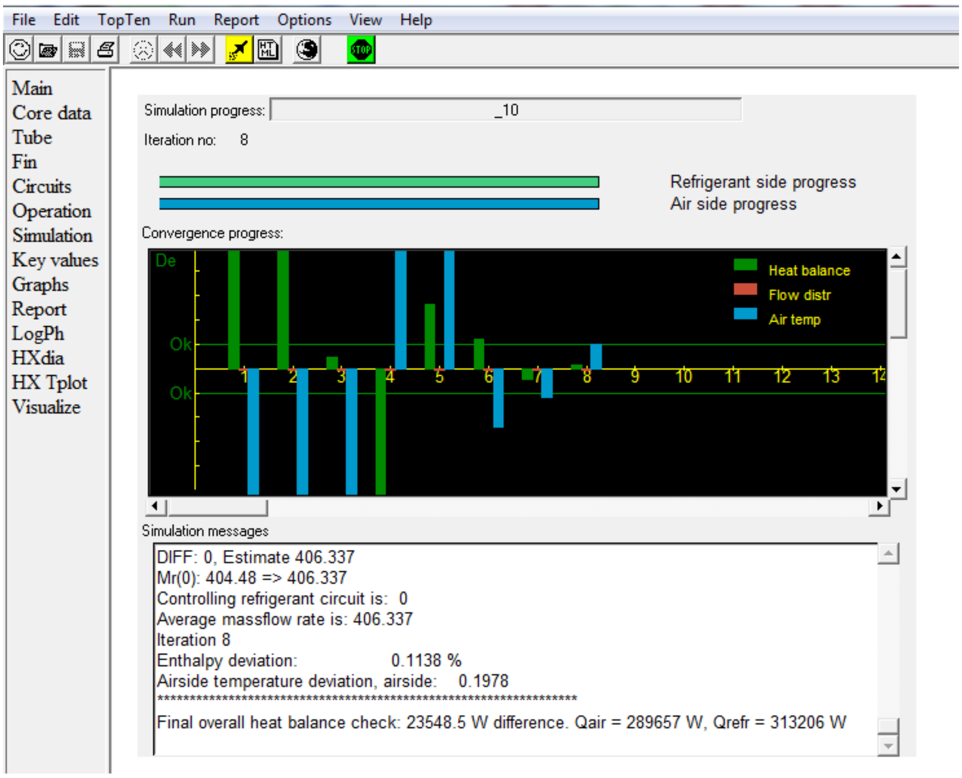
```
f=1;


A=f;
B=(0.25)/(log10((eps/(3.71*D))+(2.51/(Re*sqrt(f)))))^2;


n=0;
while round(A,4)~=round(B,4)

        f=B;
        A=f;
        B=(0.25)/(log10((eps/(3.71*D))+(2.51/(Re*sqrt(f)))))^2;

        n=n+1;


end



%Losses limit

h=0.0826*f*(Q^2/D^5)*1;
dp_section=h*ro_liq;


dpl(p)=dp_section;
p=p+1;
Diam(p)=D;
end

i=1;
while dpl(i)-dp_limit>0

    dp_section=dpl(i);
    dp_correct=dpl(i+1);
    i=i+1;

end
format long
disp(dp_limit);
disp(dp_correct);
disp(Diam(i+1));
```

```
     5.681952000000001e+03

     5.648589921755187e+03

   0.116900000000000
```

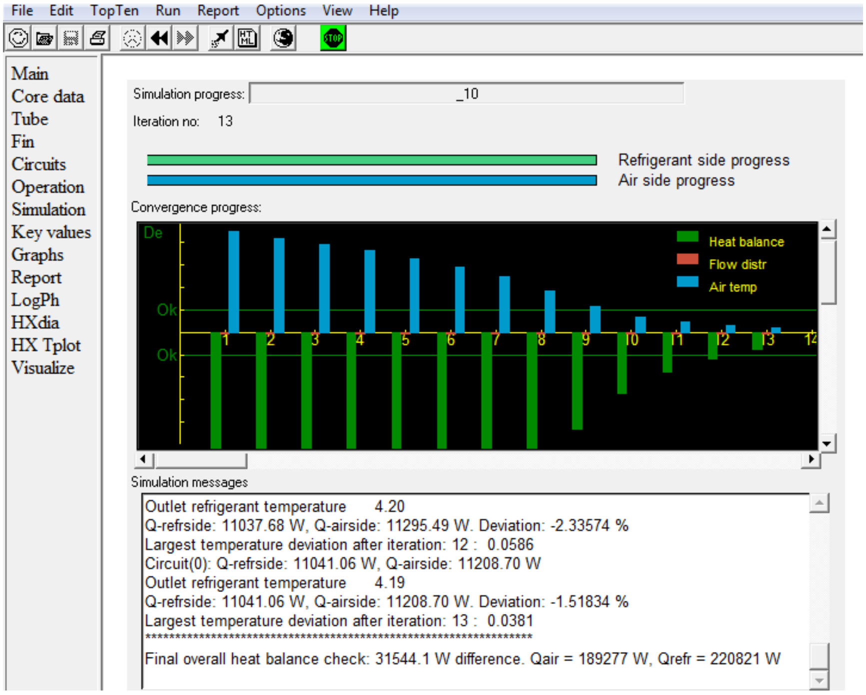
Published with MATLAB® R2016b

Annex B

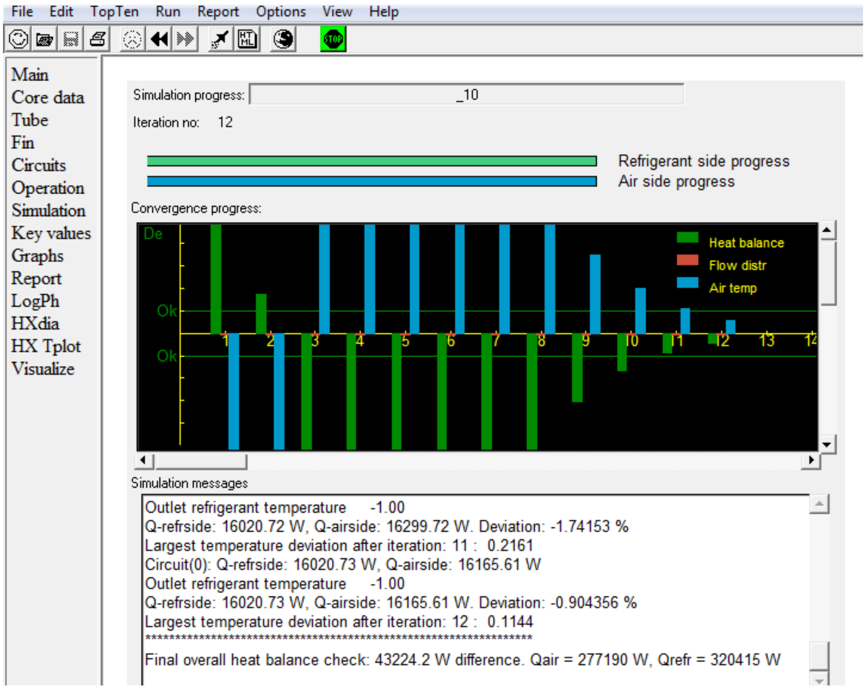
A2. Evaporator simulation results



A3. Thermosyphon module simulation results



A4. CO_2-ammonia module simulation results



A5. CO_2-CO_2 module simulation results